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■ President R. J. S. Pigott, left, who spoke at the Passenger Car and Production Meeting dinner, March 5. Dr. Robert E. Wilson, chairman of Standard Oil Co. (Ind.), dinner speaker, with Robert Insley, chairman of Detroit Section, and B. E. Hutchinson, vice-president of Chrysler Corp.

Lighter, Smoother, Sturdier, and Safer CAR IS GOAL

LESS weight, less vibration, longer life, and more safety were the car design trends detailed by example and exposition at the SAE National Passenger Car and Production Meeting in Detroit, March 3-5. New production processes and controls - to get these results and still hold costs in line - were emphasized as well.

WEIGHT SAVING SOUGHT

Engineers constantly have weight saving in mind as a subsidiary advantage when attacking other design objectives, discussions throughout the meeting indicated.

This was mentioned as a corollary to direct attacks on the vibration problem, as one objective of more extensive study of stress analysis to body structures, and as an important reason for using aluminum for car doors and deck lids, as well as for such parts as running boards, head lamp housings, and carburetor mountings.

Steel scarcity has underlined importance of substituting the lighter metal, even at an increase in cost.

Body engineers appear to be looking to more intensive application of stress analysis to get lighter bodies without reducing structural efficiency. Leading body men agreed that structural efficiency has been decreasing rather than increasing in recent years as body weight has been increased and rigidity has declined. If present rigidity is sufficient, they said, it should be maintained while designing a lighter structure.

But the body men don't think lighter structures are going to come by use of aluminum for bodies as a whole.

ELIMINATING OBJECTIONABLE VIBRATIONS

Progressive elimination of objectionable vibrations is promised as a result of new light cast by the meeting on this perennial design problem. "Threshold limits" were suggested - and quantitatively specified - in research results revealed at the meeting, as were details

of new "controlled frequency" engine mounts by which the engine, in any of its planes of vibration, is prevented from synchronizing with the frequency of any other part of the car.

LONGER LIFE FOR PARTS

Longer life for future cars was presaged in design changes proposed as a result of improved knowledge of cylinder wear and distortion growing out of new contour-gage measuring results; by satisfactory application of know-how for bonded brake linings; by increasingly reduced vibration throughout the car; and by more assured performance of body structures resulting from increasingly accurate use of stress analysis in body design.

Longer brake life is growing out of elimination of rivets in applying lining. With bonded lining, drum scoring from rivet holes loaded with dust is eliminated; the friction material can be used better because rivets are no longer there to strike drums prematurely. (The new "hot-shot" process which makes bonding practical, involves applying high temperatures and pressures to the cement for a short period and following up with an oven cure after the assembly has been removed from the clamping jig.)

Longer effective life for cylinders may come through seven proposed design changes suggested from studying results of application of new cylinder wear measuring methods. Cylinder wear and distortion may be reduced, according to the data revealed at the meeting, by: (1) elimination of tie-in between cylinder head cap screw or stud bolt bosses and the cylinder wall; (2) provision for equal coolant flow to all areas of the cylinder wall; (3) gasket design which assures least top-deck distortion from unequalized loading conditions; (4) cylinder head and valve port designs which assure symmetrical applications of combustion pressures to pistons; (5) improved maintenance of existing cylinder processing equipment; (6) maintenance of uniform cylinder wall thickness; (7) sufficient structural stability for entire engine block to resist distorting forces of engine assembly and operation.

DESIGNING FOR SAFETY

The design-for-safety theme ran through many sessions, emphasized brake and headlight trends, touched on the ability of a car to protect itself in an accident and other elements of design.

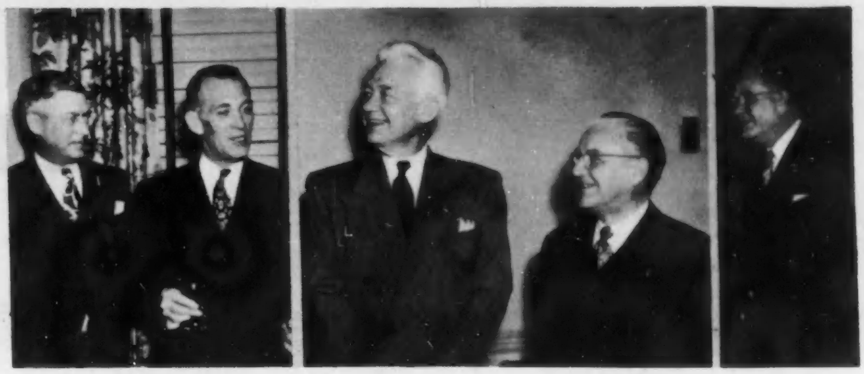
The car buyer can buy more comfort, more speed, or more beauty if he is willing to pay for them, it was brought out at one session. But if he wants more safety, he is in bad shape. Buying a higher priced car will not get him much more seeing ability or stopping ability. Emphasized was the engineers' responsibility to improve safety by designs so fool-proof that they will outwit bad drivers. Suggested to improve safety were: greater glass area, wider-area wipers, speedometers emphasizing safety instead of speed, and better detailed lighting devices.

Safety resulting from improved headlights already has been increased considerably in recent years, lighting engineers brought out. Although polaroid lighting is the only development in the offing which seems to offer substantial additional relief from glare, the industry - after studies lasting several years - is still unable to find satisfactory solutions to the problems of the changeover period. Appreciably increased cost of equipment and the nuisance of viewing the road through an analyzer are other problems which caused the industry recommendation against polaroid adoption "at this time."

PRODUCTION DEVELOPMENTS

Production thinking developed at the meeting pointed to closer coordination between manufacturing and design engineers and - by specific example - stressed economies generated by application of new welding and forging methods.

"Improved quality of product, reduction in cost, and increase in volume go hand in hand," one engineer emphasized in detailing the overall process control



George A. Delaney, chairman of the SAE Meetings Committee, left. To the right is SAE Vice-President J. W. Greig of the SAE Body Activity, Vice-President A. W. Frehse, of the Passenger Car Activity and Vice-President J. B. Armitage, of the Production Activity. At the right is G. B. Allen, who served as general chairman

methods used in one of the industry's largest operations. Thermal bonding of brakeshoe linings to the shoes, another said, is pointing the way to bonded lining which will last the life of the car. Experience gained in handling aluminum in the last 10 years, still another mentioned, led to engineering of completely new engine mounts which, in turn, permitted weight reduction. Many other production-design relationships were outlined as well.

Economies detailed centered largely on the welding and forging areas. Poke welding, for example, - a new timesaving method - was stressed as useful for boxed-in sections at pillar reinforcements to the floor pan, seat risers to the floor pan and similar locations. (This new process involves use of a light welding gun, a fixed electrode, a device for cooling the electrode, and a means of surrounding the weld area with argon gas to prevent oxidation.)

Examples showing fantastic material savings by forging as against other processes were cited in a review of near-finish-size forgings. Three and a half parts are being made from the same amount of material which formerly produced only one in one instance mentioned. In another, substituting forging and thread rolling in making wheel nuts weighing 6 oz when finished, about 12 oz of metal was saved. . . . By combining forging and rolling operations, it was said, marked economy is obtained in material used and more parts per hour than could be achieved by conventional machining and turning operations for producing pinion gears, car seat regulator pads, automobile tie rods, camshaft forgings, wrenches, steering spindle arms and other parts.

Petroleum economics, which conceivably could affect many basic elements of car design and production, got the spotlight at the dinner which closed the meeting when Robert E. Wilson, Standard Oil of Indiana board chairman, said that prospects of sufficient gasoline, diesel fuel, and fuel oil for the remainder of this year and 1949 are dim.

Overall U. S. figures show demand this year was 14% over supply:

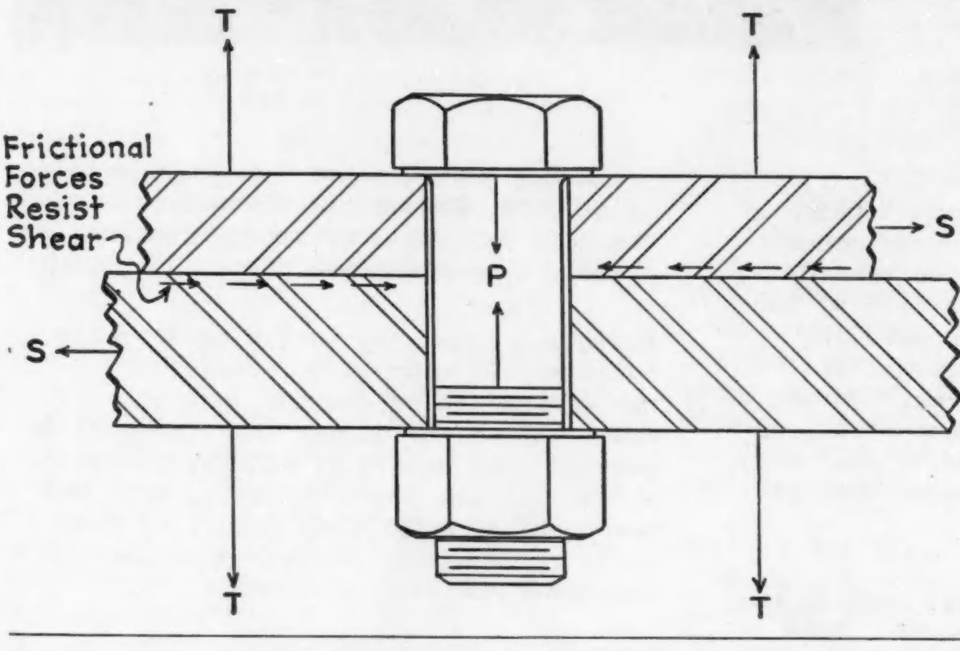
Gasoline	9%
Kerosene	39%
Distillates	24%
Residual fuels	24%
Other products	11%

A five or six percent increase in production capacity next year is probable, he reported, but there are no indications of a slackening of demand in view of increased mileage of cars and trucks, more diesel locomotive mileage, more home heaters, and other demands upon available refining capacity.

SAE President R. J. S. Pigott agreed with Dr. Wilson in stating that no one should believe the prophets of doom who say that our oil reserves will be exhausted within 10 years.

B. E. Hutchinson, Chairman of Chrysler's Executive Committee was toastmaster, while the dinner preceding was opened by Detroit Section Chairman Robert Insley.

Preloaded Bol



■ Fig. 1 - Bolt preload, P, resists both tensile and shear forces on the structure

(This paper will be published in full in SAE Quarterly Transactions)

PRELOADING a bolt and maintaining that tension throughout the life of the joined part helps the bolt perform its clamping function better and makes it last longer.

Fig. 1 shows behavior of preload on a bolt holding two pieces together. Preload tension, P, in the bolt shank is developed by torquing the bolt head or nut or in other ways. As long as tensile forces, T, on the bolted structure don't exceed P, the parts will not separate nor will the bolt extend. Tension in the bolt doesn't increase; it remains at its preload value P.

While not strictly accurate, this holds for all practical purposes if the bolted material is well fitted and rigid compared with the bolt. If the bolted material is elastic - the case with gasket material - bolt load will exceed preload as tension is applied to the structure.

Preloading also is useful in shear loaded joints. Despite shearing forces, S, in Fig. 1, the joint doesn't slip, so long as S is less than P, because of frictional resistance between the parts produced by preload tension. Since the joint doesn't slip, there's no shear on the bolt and no change in load from initial preload value. Another advantage of such connection is the transfer of stress by friction from one plate to the other at areas remote from the hole. This prevents high stress concentrations near the hole.

Preloading bolts under tensile loadings also resists fatigue. Almen[1] shows that endurance life of a bolt in comparatively rigid parts increases rapidly as the

1. See SAE Transactions, Vol. 52, October, 1944, pp. 151-158; "On Strength of Highly Stressed, Dynamically Loaded Bolts and Studs, by J.O. Almen.

Betters Design

BASED ON A PAPER* BY **W. C. Stewart**

Technical Advisor, American Institute of
Bolt, Nut and Rivet Manufacturers

bolt is progressively tightened. See Fig. 2. It becomes infinite as initial bolt tension approaches maximum bolt load.

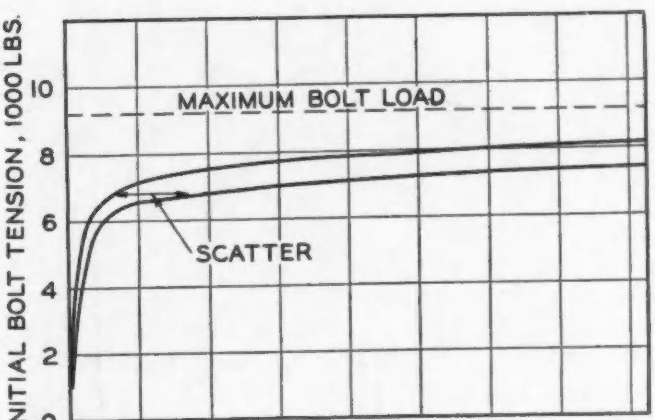
This happens because increased preloading reduces amplitude of stress change in the bolt to a safe value - if the bolted material is rigid. The bolt's fatigue resistance depends solely on preloading. Less rigid bolted material allows a stress cycle of greater amplitude and leads to early failure. Such condition calls for special attention to factors reducing stress concentration, such as proper filleting under the bolt head, surface finish, contour and finish of thread roots.

Rolling thread roots and rolling and cold-forming threads also help. Rolling's triple action nearly doubles endurance limit. It gives a smoother finish; increases the material's tensile strength and sets up compressive stresses in the root. Heat-treatment after rolling undoes the cold-working good and compressive stresses; but such threads usually are superior because of their better finish.

By minimizing stress-change amplitude in the bolt, proper preloading imparts a locking effect. It prevents nuts and bolts from shaking loose and unscrewing, as they tend to do when stress changes vary too much.

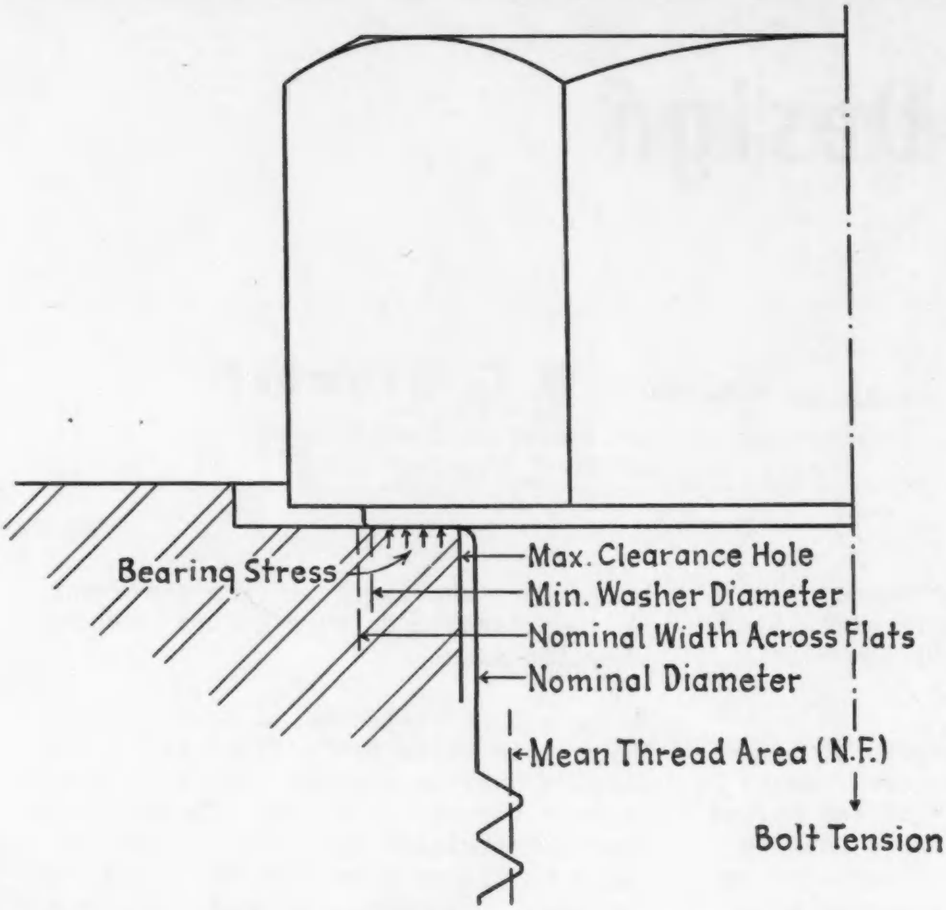
Bolts can be preloaded in several ways. Very large bolts are drilled along the axis, a heating element inserted, then lightly torqued in place. Thermal contraction preloads the bolt. Sometimes large bolts are stretched with a hydraulic jack. Releasing the jack transfers the load to the bolt. Power devices also are available for tightening. Most usual method is the ordinary hand wrench, often equipped with a torque-measuring device.

When recommending a specific torque, as in production work, the designer should realize that friction consumes at least



■Fig. 2 - Bolt life depends on its initial preload tension

*Paper "Applications, Materials, and Specifications of Bolts," was presented at SAE Annual Meeting, Detroit, Jan. 12, 1948.



■Fig. 3 - Without enough contact bearing area, bearing stress induced by bolt tension may crush bolt or bolted-part material

90% of applied torque. Torque to produce a given tension in a bolt equals the torque coefficient times nominal bolt diameter times bolt tension. With inch and pound values, the torque coefficient is about 0.2. But it varies somewhat with bolt size and friction condition.

It's equally important to maintain preloading during the life of the bolted assembly. Bolt tension probably relaxes some in service. It may come from flattening of contact-surface rough spots or plating, or from extrusion of dirt, paint or plating from between contact surfaces. Also metal at high stress-concentration points may flow. Another cause may be crushing of the bolted material under the bearing face or crushing of bolt or nut. Crushing depends on the net bearing area in contact and relative physical properties of bolt and bolted material. Bolt head and nut should be wide enough, with maximum clearance hole, to make sufficient contact so that fastener material will not crush under maximum bolt load. See Fig. 3.

As mating surfaces adjust themselves, relaxation stabilizes at a bolt tension less than initial preloading. That's why a big enough bolt should be used where preloading is critical. It can be preloaded beyond requirements in anticipation of relaxation.

Since loss of bolt grip induced by relaxation is generally fixed, relaxation effects are more important in short than in long bolts. (Long bolts are assembled with more compensating elastic elongation.) You can get more elastic elongation by "waisting" the unthreaded portion of the bolt, with a patented flexible head, or by elastic deformation of the threads.

Much more needs to be known about bolt behavior under different conditions of use. A professional engineering society in the mechanical field should sponsor a research project to compile present knowledge and to solve unanswered problems.

(Complete paper on which this article is based is available from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

BUICK'S DYNAFLOW DRIVE

First American Self-Shift Transmission

Featuring Hydraulic Torque Converter

EXCERPTS FROM A PAPER* BY

Charles A. Chayne

Chief Engineer, Buick Motor Division, GMC

(This paper will be published in full in SAE Quarterly Transactions)

BUICK'S Dynaflow Drive fulfils the designer's dream of an automatic transmission offering optimum performance at all speeds, that automatically selects a continuously variable range of torque ratio with complete driver ease.

The Dynaflow, Fig. 1, primarily consists of a hydraulic torque converter and, secondly, a planetary-type gear train that provides a reverse and two forward driving ranges. There is no clutch pedal for disconnecting the engine. The driver selects driving ranges by a lever on the steering column, not unlike the conventional gear shift lever. See Fig. 2. All he must use to operate the car for practically all normal driving are accelerator, brake, and steering wheel.

Driver operation of a Dynaflow car is extremely simple. He starts the engine with the selector in "P" (Park) or "N" (Neutral), both starting positions. He chooses the desired range by moving the selector, and gets the car under way by pressing the accelerator. Virtually all operation of the car can be done in Drive range, from traffic driving to transcontinental touring. Of course, the driver must use Reverse for maneuvering. Occasionally it's desirable to use Low for descending a grade, extended heavy pulling at low speed, or emergency application of extreme torque.

Shifts from Low to Drive may be made at any speed and throttle position merely by moving the selector, nothing more. Shifts from Drive to Low may also be made at any throttle setting, but are not advisable above 40 mph - among other reasons because of severity of engine braking at higher speeds. If it should become necessary to "rock" the car out of a tough spot, the driver can do it easily by applying light throttle and merely moving the selector between Reverse and Low at the desired frequency.

Simplicity of operation is made possible by the hydraulic torque converter, which transmits the total engine output under all conditions of operation.

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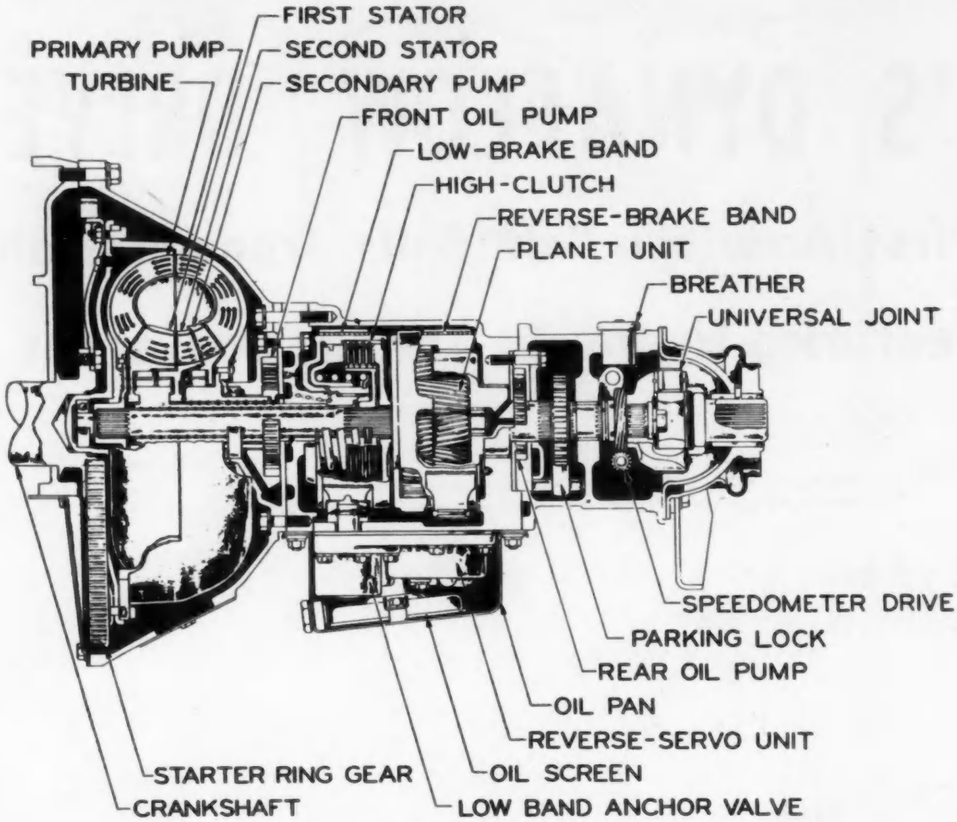
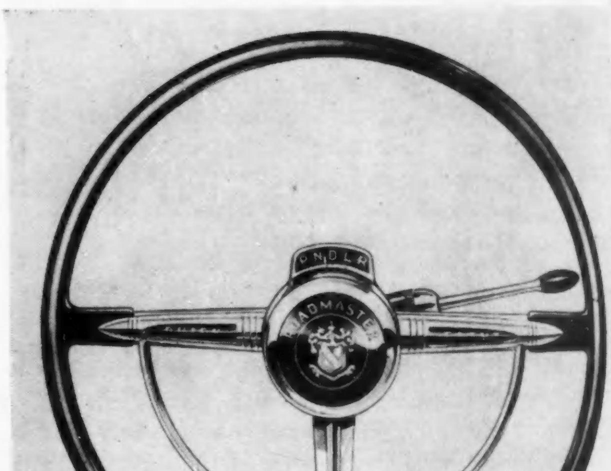
* Paper "The Buick Dynaflow Drive," was presented at SAE National Passenger Car and Production Meeting, Detroit, March 5, 1948.



■ Fig. 1 - Cutaway view of Buick's Dynaflow Drive

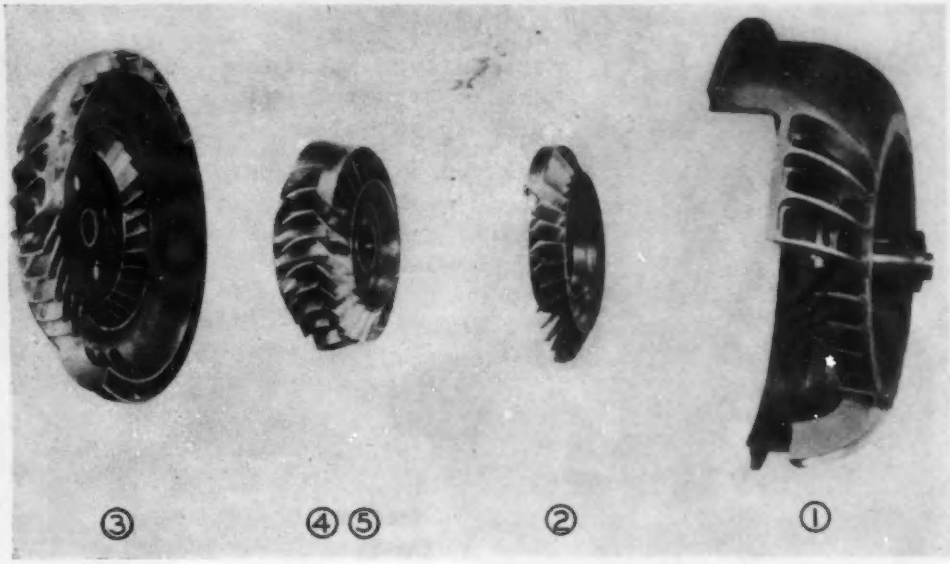
The converter consists of five annular sections, shown in Fig. 3. Each section consists of inner and outer shells enclosing curved vanes that provide impulsion of the contained fluids. These various "wheels" are (1) the primary pump - driven by the engine; (2) the secondary pump - mounted by means of a free-wheeling clutch upon the primary pump hub - this secondary pump is permitted to rotate about its axis faster, but never slower, than the primary pump; (3) the turbine, or driven member, which is splined to the turbine shaft and drives the car; (4) the first stator; and (5) the second stator, both of which are reaction members. Each stator is mounted on a free-wheel clutch, permitting motion only in the direction of engine rotation. The hub of these individual free-wheel clutches is splined to a reaction shaft, rigidly attached to the transmission case.

Clearest description of the converter's functions comes from examining its action through a normal sequence of power application.



■ Fig. 2 - The selector lever on the steering column operates the hydraulic control system and a parking device. There are five positions on the quadrant: P (Park), N (Neutral), D (Drive), L (Low), and R (Reverse). Settings can be made by feel. Each position has a detent which holds the selector in place

Fig. 3 - Torque converter components are: (1) engine - driven primary pump, (2) secondary pump (3) turbine, (4) first stator, and (5) second stator



Assuming a standing start with the engine at idle, the primary pump slowly turns with the engine, and the turbine is stationary. Centrifugal force on the oil in the pump makes the fluid flow outward, and overcomes the resistance of oil in the stationary turbine and other members. A gentle circulation is present - outward through the pump and inward through the turbine. The engine is idling, although "connected" to the power train. And because of the low circulation rate, no appreciable torque is transmitted.

Now, to accelerate, the throttle is opened. Designed capacity characteristics allow the engine to speed up and rapidly approach its torque peak. Oil in the engine-driven pump is now moved with high centrifugal and angular momentum, circulating with extreme rapidity through the stationary turbine. Oil projected from the pump into the turbine is altered in direction by the turbine blades and exerts a direct force upon the turbine.

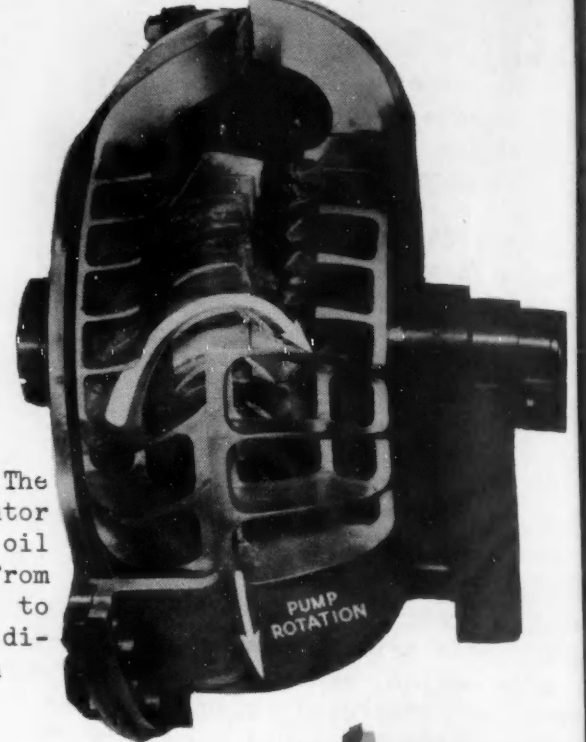
Maximum alteration of oil stream direction through the turbine is accomplished by a reverse curvature of turbine blades.

Oil passing through the turbine is discharged in a direction opposite to the rotation of turbine and pump, as in Fig. 4. When the turbine is held stationary, this discharged oil stream has a speed greater than its speed when projected from the pump. Speed of the oil mass increases because the oil is deflected by the stationary turbine blades without removal of energy, except for very small friction losses. When returned toward the axis by the physical confines of the turbine, the oil enters passages of decreased sectional area between the turbine blades. Since the amount of oil leaving must equal that entering the turbine, it becomes clear that under the conditions described, discharge takes place at increased speed.

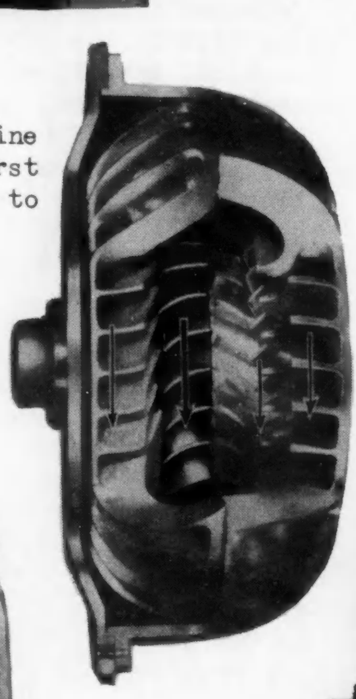
Velocity of the oil leaving the turbine is negative in direction with regard to pump rotation; but with little manipulation it can be converted to a positive velocity and added to pump output. This conversion is achieved by interposing a static member in the turbine discharge with curved blades to redirect the oil stream into the direction of pump rotation. See Fig. 5. Engine torque applied to the pump is capable of accelerating oil from rest to a given pump output velocity. If oil is supplied to the pump input with a positive velocity relative to pump rotation, pump output increases directly and becomes the sum of input velocity and output velocity due to engine torque alone.

Since force applied to the turbine bears a direct relationship to the oil mass projected, it is apparent that torque on the turbine has increased in the same ratio as the increase in pump output, al-

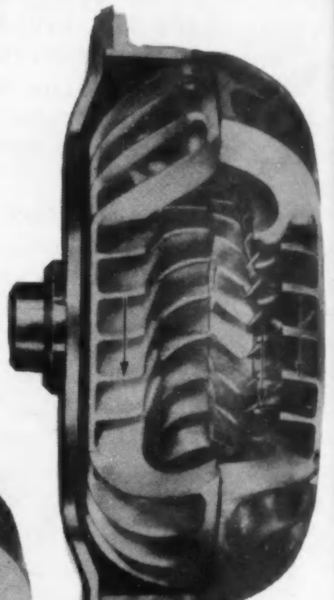
■ Fig. 4 - Oil is discharged from turbine against turbine and pump rotation



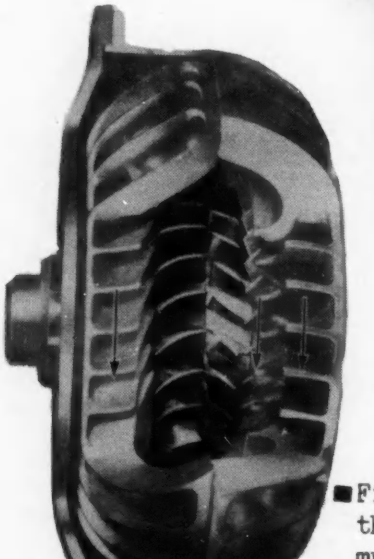
■ Fig. 5 - The first stator reverses oil flow from negative to positive direction



■ Fig. 7 - As turbine speeds up, first stator starts to free wheel



■ Fig. 9 - In the second stage of operation both stators are free wheeling and the converter acts as a fluid coupling



■ Fig. 6 - Torque on the turbine, now multiplied, puts the car in motion. Stators are still held stationary

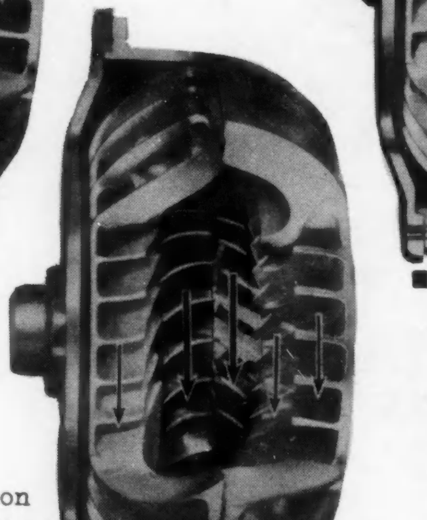


Fig. 8 - Now second stator starts to free wheel

though engine torque has remained the same. Note at this point that the static members exert a large force upon the oil stream, but virtually no energy is dissipated in redirection since they are prohibited from moving under reaction force. Therefore, they are unable to extract work from the moving oil.

Degree of mechanical advantage obtained depends on the ability of the stators to redirect the oil into the pump at the proper angle - a function of stator blade curvature. In the Dynaflow converter, effective speed of the oil redirected into the pump by the stators is 1.25 times the pump output rate due to the engine torque application alone. This combination increases torque on the turbine by a factor of 1 plus 1.25 or 2.25. Therefore maximum torque multiplying ability of this unit is 2.25 to 1.

Returning to the sequence of power application, let's assume that torque applied to the turbine output places the car in motion by rotating the turbine and its output shaft, shown in Fig. 6. Maximum acceleration is still demanded by holding the throttle open. As the turbine picks up in speed, so does the engine, maintaining a fairly large difference in speed between pump and turbine. Circulation continues at a high rate; large reaction forces are present; torque multiplication exists.

As turbine speed increases in what we termed a positive direction of rotation, negative velocity of turbine discharge is reduced by an amount equal to the forward or positive speed of the turbine. Because the turbine blades are running away from the stator blades (which reduces rearward discharge velocity against the stators), the speed at which oil is directed into the pump diminishes. The pump output rate, and consequently torque multiplication, decreases. (This condition is further augmented by increase in engine speed, which causes the pump blades to move away from the stator blades more rapidly.)

Because of changing speed of rotating members and consequent variations in oil stream velocity through the converter, the angles at which oil enters upon the blades of converter components have undergone considerable change. This is true with one notable exception. Effectiveness of converter blading depends on maintenance of proper entrance angle; therefore, compensation must be made to preserve efficiency over the wide range of speeds required. It's done by breaking up fundamental converter elements into suitable segments, and mounting these so that they are used when required, and removed from action when not required. The Dynaflow converter has two stator wheels and two pump wheels. Free-wheel clutches within the first and second stators and the secondary pump provide the desired action of their supported members.

As turbine speed increases, oil discharged from it has a reduced rearward motion. A condition is reached where this rearward motion is so diminished that the oil's angle of entry into the first stator is no longer satisfactory. (Fig. 7). At this point reactive force no longer acts on the blades of the first stator. In fact, the oil is beginning to impinge on the "back" side of its blades, causing the first stator to free wheel in the oil stream so that it presents minimum obstruction to flow. The second stator at this time is still being entered effectively because of the different location of its blades. It continues to redirect oil into the pump. Torque multiplication is reduced but still present.

With continued increase of car speed demanded, turbine speed rises to a point where oil is discharged with less rearward speed than the forward speed of the turbine (Fig. 8). Positive direction of flow now brings about free wheeling of the second stator. There is no longer any redirection of oil entering the pump; torque multiplication has ceased.

It is readily conceived that a change of entrance angle into the pump occurs simultaneously with the variation of velocity within the stators (Fig. 9). Under the condition where the turbine is stationary, oil at high speed is redirected into the pump by the stators. Its entrance angle to the blades of the primary

pump is optimum for impelling the intense oil flow. At this time, the oil stream from the stators travels in a direction which moves the more obliquely-bladed secondary pump freely upon its clutch.

However, as regenerative action through the stators decreases with increasing turbine speed, oil approaches the pump with declining velocity until the circulation rate becomes less than that which the pump can produce through application of engine torque alone. As this comes about, the secondary pump slows down until it no longer overruns the primary pump on which it is mounted. The secondary pump now is carried forward by the primary pump hub, to which it has locked and, by virtue of its more oblique blades, enables efficient collection and impulsion of oil by the pumps.

Exception to this condition of entrance angle variation is the junction between pump and turbine. Here, high oil circulation together with great speed difference between pump and turbine (lower circulation exists at lower speed difference) produce a constant resultant entrance angle of the oil stream into the turbine through the range of converter operation.

CONVERTER TURNS COUPLING

With the stators free wheeling and the pump turning as a unit, the converter now functions as a fluid coupling transmitting torque at a 1 to 1 ratio. Sufficient residual speed differential remains to enable transfer of oil with higher momentum from the pump to turbine, where it gives up energy, displaces toward the axis, and is again pumped.

Torque multiplication varies smoothly with throttle demand for power application. Because of inherent mechanics of the converter, speed differential is greater at lower car speeds; but performance characteristics remain the same. This condition is entirely desirable because it permits relatively free idling of the engine, and allows acceleration of the engine to a speed productive of maximum output torque at the instant-of-starting condition. For optimum performance, the converter must be tailored to the particular installation.

Performance curves of the Dynaflow converter, Fig. 10, clarify the rather intricate relationships of speed, speed differential, torque multiplication, and relative efficiency. Points of inflection on the efficiency-versus-output-speed curve indicate the multiphase operation of the unit. These "kinks" show when the transition occurs between stator phases. Comparison of this efficiency curve with that of a typical rigid-stator converter demonstrates the marked increase in efficiency of the Dynaflow converter especially at higher speeds. Comparison of this curve with that of a conventional fluid coupling is also extremely interesting in the lower and middle ranges, especially since the converter provides increased output torque which the fluid coupling alone is unable to achieve.

The question is already forming in your minds: "Yes, all that is very interesting - but how can these efficiency curves stand up against those of conventional gear boxes?" The most adequate and direct answer is that the overall operating efficiency of complete cars with and without Dynaflow is practically the same.

Dynaflow

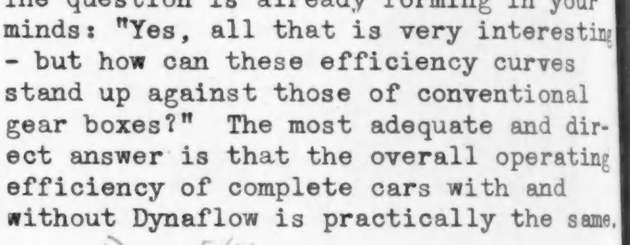


Fig. 10 - Torque converter performance curves

It must be borne in mind that a conventional transmission forces engine operation into a state of confused and materially reduced efficiency; the Dynaflow allows engine operation at speed ranges productive of highest efficiency.

This compensation between engine and transmission works out about equally, while the converter adds the advantages of infinitely more accurate power application and ease of driving. These latter items are not measurable in terms of mere efficiency curves. With the engine no longer constrained by mechanical connection to operate at the "speed-of-the-car" gentle yet flashing getaway is presented, without jars or jerks of gear shifting. At the same time, delicate control of full pulling power is available for inching out of tough spots. The measure of this unit must be made in terms of "ability to do work - conveniently."

To provide a "super-low" for extremely hard pulling and for engine braking on descent of heavy grades, as well as to provide for necessary reversal of direction, a planetary gear set is provided in the Dynaflow Drive. This gear set is controlled by a manually selected hydraulic system. The planetary is a dual pinion type set, illustrated in Fig. 1, driven by the turbine shaft.

In order of their appearance in the power train the gears in the set are: The reverse sun gear splined to the turbine shaft. Meshed with the input sun gear and mounted in a planet carrier are three low planet pinions. These low pinions in turn mesh singly with three reverse planet pionions, also mounted in the planet carrier. The reverse planet pinions mesh with both an external tooth sun gear called the low reaction gear, and an internal tooth reverse ring gear. The low reaction gear is mounted on a large plate which is integral with the driving range clutch; the turbine shaft passes freely through its center. The driving range clutch is a hydraulic-piston type, multiple-disc wet clutch which, when engaged, secures the low reaction gear to the turbine shaft through the clutch hub, locking up the gear set, producing direct drive.

Low range is secured by holding the low reaction gear stationary with a brake band applied to the outside of the clutch drum, with the clutch, of course, released. Reverse is secured by holding the reverse ring gear stationary, again by means of a brake band, with all other controls released. With all controls released, the gear set is in a neutral condition. In both low range and reverse an equal gear reduction is present (1.82 to 1). With torque multiplication of the converter, this provides a maximum transmission torque ratio of about 4.09 to 1.

Judged in its entirety, the Dynaflow transmission offers for the first time a degree of flexibility, ease, and performance never previously presented. It makes car operation safer because of simple operation and instantaneous response. The driver can concentrate more on surrounding conditions and maintain absolute toe-tip control of power application.

(Complete paper on which this article is based is available from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

You'll Be Interested To Know...

SAE members are invited to participate, by attendance or presentation of papers, in celebration of the centennary of "The French Engineer" in Paris, May 28 to June 8. Full information is available by writing to O. Yadoff, Association des Ingenieurs-Docteurs de France, 934 Fifth Ave., New York, N.Y.

* * *

Recently proposed revisions to the SAE Constitution (see SAE Journal, September, 1947) have been approved by mail ballot of the membership.

(This paper will be published in full in SAE Quarterly Transactions)

How to Predict TRUCK PE

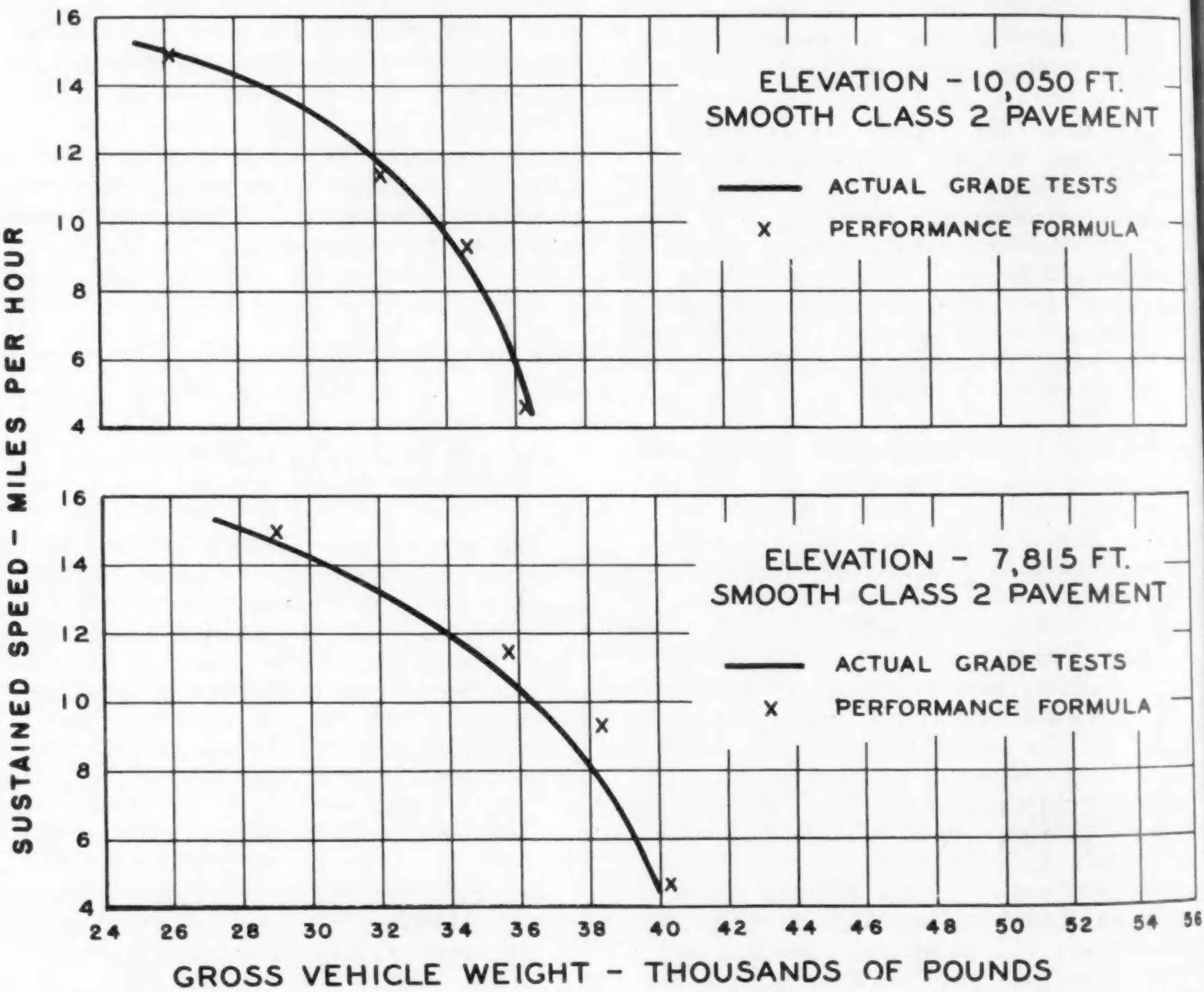
FLEET men are beginning to uncover ways of matching truck performance to operation requirements. Despite controversies and lack of sound experimental data, formulas are being developed for computing quite closely the hill-climbing ability and grade-ability of the truck.

A grade-ability formula that shows up well when compared with actual road performance is:

$$GVW = \frac{\left[NHP_e - (FHP + AHP)\right] 37{,}500}{\frac{(7.6 + 0.09V + C + g)\, V}{10}}$$

where:

GVW = gross vehicle weight in lb,
V = speed in mph,
NHP_e = net hp available at the given elevation,
FHP = chassis frictional resistance hp,
AHP = air resistance hp,
g = grade in percent,
C = increment for surface type.



■ Fig. 1 - Comparison of hill-climbing ability determined by grade tests and performance formula

BASED ON A PAPER* BY **Carl C. Saal**
Highway Engineer
Public Roads Administration

PERFORMANCE

Proximity of calculated results and performance in actual road tests, in Fig. 1, speaks well for this formula. These tests at four different altitudes were conducted by the War Department with a 3-axle tractor semitrailer. They demonstrate the reliability of the formula for different altitudes and surface types.

Linear accelerations can be computed from the following expression:

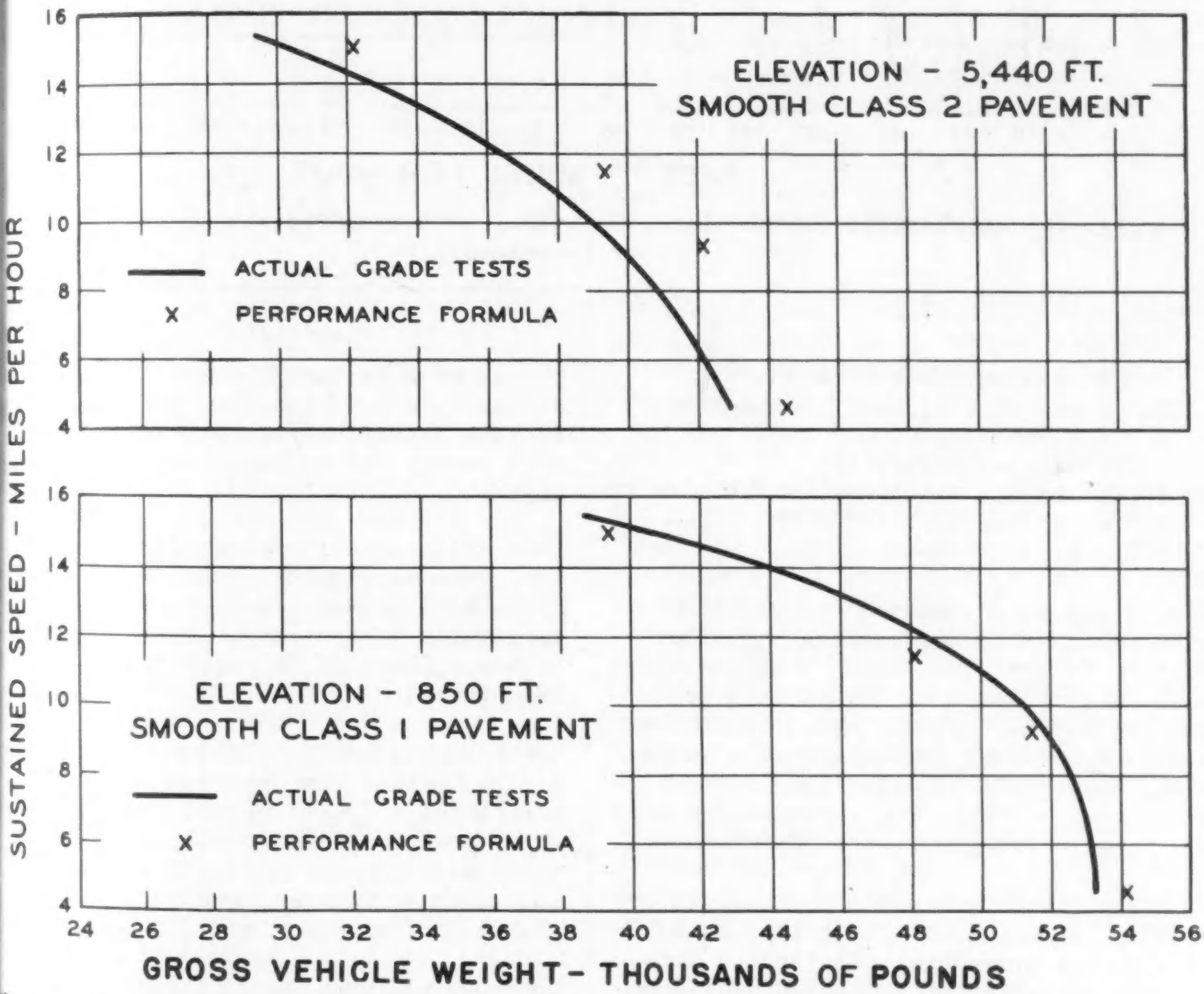
$$a = \frac{\left[NHP_e - (AHP + FHP + RHP + GHP)\right] 375}{(m + k)\, V}$$

where:

a = linear acceleration in ft per sec ^{2},

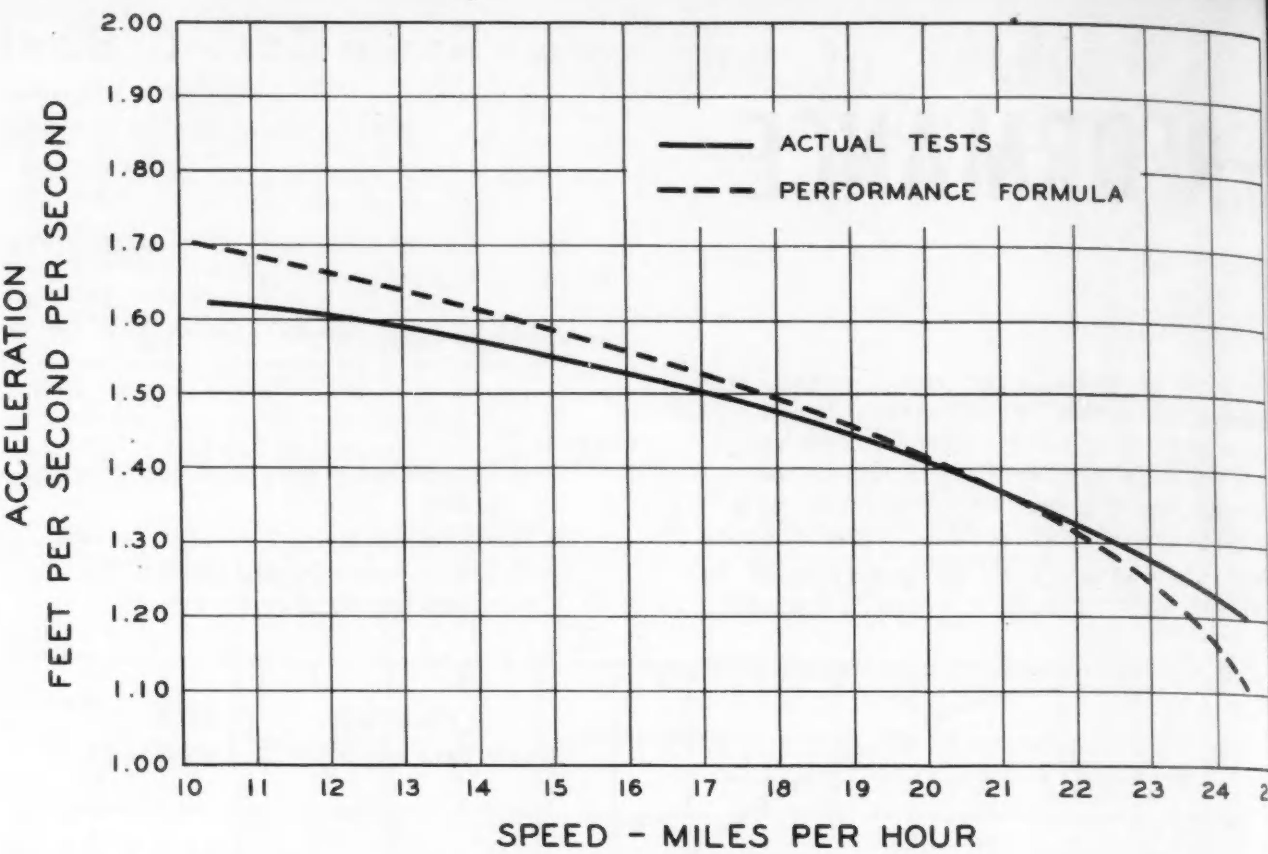
V = speed in mph,

m = gross vehicle weight in lb divided by 32.2,



The vehicle is a 3-axle tractor semitrailer, in third gear, on a 6% grade at various altitudes

*Paper "An Evaluation of Factors Used to Compute Truck Performance," was presented at SAE Annual Meeting, Detroit, Jan. 13, 1948.



■ Fig. 2 - Acceleration on the level in actual test compared with performance-formula values

k = mass equivalent constant,
NHP_e = net hp available at the given elevation,
AHP = air resistance hp,
FHP = chassis frictional resistance hp,
RHP = rolling resistance hp,
GHP = grade resistance hp.

Fig. 2 compares computed accelerations with actual road performance. The test vehicle in this case was a 3-axle tractor semitrailer loaded to 26,000 lb gross vehicle weight. Actual acceleration was observed on level smooth concrete pavement with the vehicle in third gear. These results also affirm the reasonableness of the formula for predicting performance.

At first glance the whole performance-prediction job may look like a simple, well-established procedure. It isn't. It's the determination of the horsepower-consuming factors in the formulas that still involves intelligent guessing and makes for differences of opinion. These truck-performance factors are: rolling resistance, air resistance, grade resistance, chassis frictional resistance, vehicle mass energy and stored energy of rotating parts.

Most reliable expressions for these factors are given in Fig. 3. These were used in grade ability and linear acceleration computations discussed above. But pet values of many truck engineers differ from these.

For example, at least three different empirical values are proposed for rolling resistance - 10, 12.5, and 15 lb per 1000 lb of gross vehicle weight. Certain investigators found rolling resistance increases with speed; yet it's common practice to assume that it's constant.

Controversy over computation of air resistance centers about the coefficient of air resistance. (In the formula in Fig. 3 it is 0.00250.) Five truck manu-

Rolling Resistance Horsepower:

$$RHP = \frac{(7.6 + 0.09V + C)\ GVW \times V}{375{,}000}$$

Air Resistance Horsepower:

$$AHP = \frac{0.0025\ AV^3}{375}$$

Grade Resistance Horsepower:

$$GHP = \frac{GVW \times g \times V}{37{,}500}$$

Chassis Frictional Resistance Horsepower:

$FHP = 1.0 + 0.00195$ RPM (Light vehicles)

$FHP = 1.0 + 0.00420$ RPM (Medium and heavy vehicles)

Horsepower Available at a Given Elevation:

$$NHP_e = (1.00 - 0.00003E)\ NHP$$

Stored Energy of Rotating Parts:

$$k = \left[\frac{I_1}{r^2} + \frac{\overline{GR}^2\ I_2}{r^2}\right] \times \frac{1}{32.2}$$

where:

GVW = gross vehicle weight in lb,

v = speed in mph,

C = increment of unit rolling resistance for surface type and condition,

A = projected frontal area in sq ft,

RPM = engine speed in rpm,

E = elevation in ft,

NHP = certified net hp,

I_1 = moment of inertia of wheels and tires (assembled) in lb-ft^2,

r = loaded wheel radius in ft,

GR = total gear reduction, and

I_2 = moment of inertia of crankshaft, flywheel, and clutch in lb-ft^2.

■ Fig. 3 - These performance-factor formulas produce fairly reliable results

facturers prescribe five different constants - ranging from Mack's value of 0.00180 to International Harvester's 0.00500 for the worst condition. Until someone performs wind-tunnel tests on full-scale commercial vehicles, engineers will have to rely on arbitrary values.

Investigations of altitude effects on gasoline engine power output have yet to yield relationships worthy of standardization.

A more satisfactory method of computing chassis frictional resistance - frictional losses in power transmission from clutch to wheels - must be found. Engineers used to figure it as a percentage of horsepower at clutch delivered to driving wheels. More recently these frictional losses have been expressed in terms of horsepower requirements at given road speeds. Limited experimental data, obtained by the Public Roads Administration, seem to need further verification.

These and other factors in need of clearer definition call for additional research to advance the problem from the realm of compromise.

(Complete paper on which this article is based is available from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

NONMETALLIC Fuel Tank Are

BASED ON A PAPER* BY

John E. Lindberg, Frank R. Zerilli, and C. R. Ursell

PAN AMERICAN AIRWAYS, INC.

OF the three basic types of aircraft fuel tanks, only two - the integral and the bladder types - are widely used in modern commercial transports.

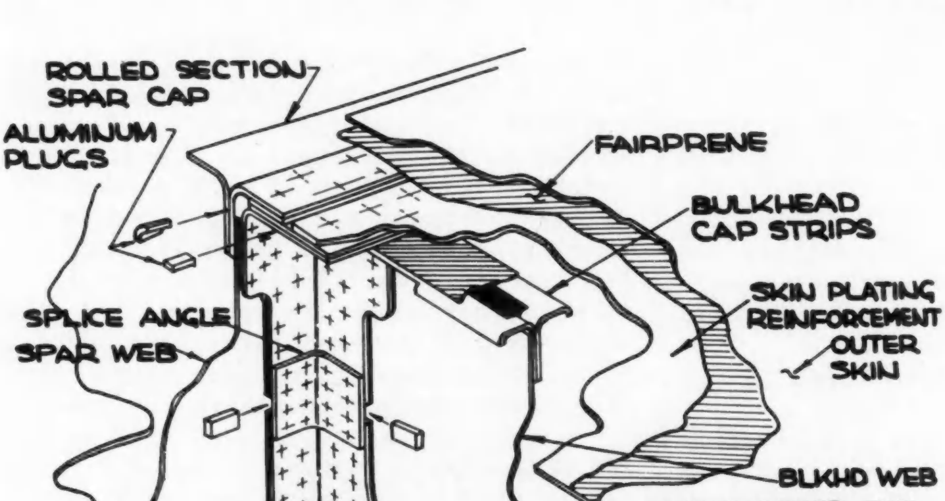
In this country, metal cells are used mainly for small aircraft, since they require almost no maintenance. They are unsuitable for long-range transports, which need tremendous fuel capacity, however, because they require larger structural openings for installation, are heavier, and provide least fuel capacity.

Integral tanks - those spaces in an airplane that utilize structural components to create fuel-tight compartments - can be classified according to sealing methods:

1. The caulked tank, which uses sealant between the faying surfaces of the structure surrounding the fuel cell.

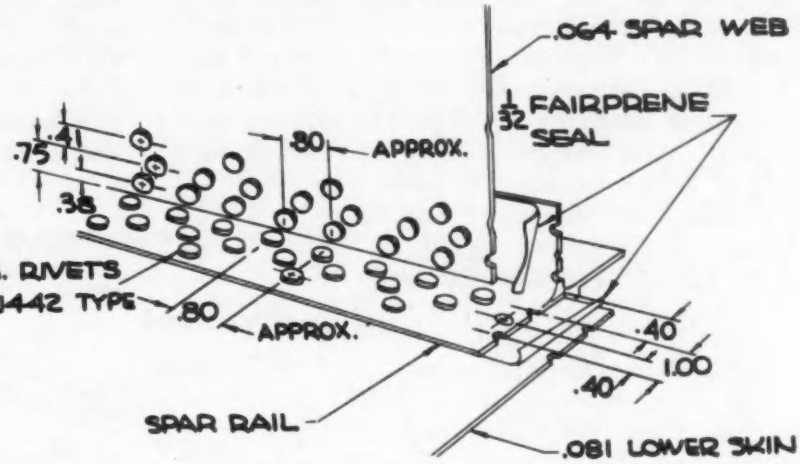
2. The adhesive membrane type, which depends for fuel tightness on an adhesive covering over part or all of the interior

* "Fuel Tanks - Integral versus Bladders versus Metal Cells," was presented at the SAE National Aeronautic Meeting, Los Angeles, Oct. 2, 1947. Discussion digested from remarks by H. E. Hjorth and O. A. Wheelon, Douglas Aircraft Co., Inc.; A. S. Baker, Lockheed Aircraft Corp.; and Fred Morris, American Airlines, Inc.



■ Fig. 1 - Fuel tank for Catalina

Are Favored for Aircraft



■ Fig. 2 - Convair 240 integral tank spar web and cap design

of the structure. The seal depends on adhesion of the membrane to the structure.

Experience with both types has been highly variable, depending on details of design, workmanship, and materials used, although so far the caulked type has had the better record. New improvements, particularly in adhesive sealants, may change this picture in the near future.

The caulked type has been extremely successful on Consolidated PBY Catalinas (Fig. 1) and in the hydrostabilizers of the Boeing B-314. The few leaks experienced were readily repaired - and the B-314's accumulated an average of 18,000 hr before they were retired.

As a result, Convair is applying PBY tank design principles to their model 240 plane, as shown in Figs. 2 and 3. The only basic change is the provision of a one-piece extruded spar cap, which eliminates the void in the two-piece formed spar cap shown in Fig. 1. Dagger fittings also carry the stress through the end bulkhead, which provides grommet seals around the structural attachment bolts. Unfortunately, some manufacturers have encountered so much trouble in attempting to make integral fuel tanks, that they have concluded it is not possible to build a sound wing if it incorporates sealants in the structurally loaded integral tank joints. They say that flexing of the wing in operation would eventually cause deterioration at the tank joints so sealed, resulting in leaks and unsound structure. They also feel that the fuel will, in time, attack the faying surface sealant material, making it unsealable.

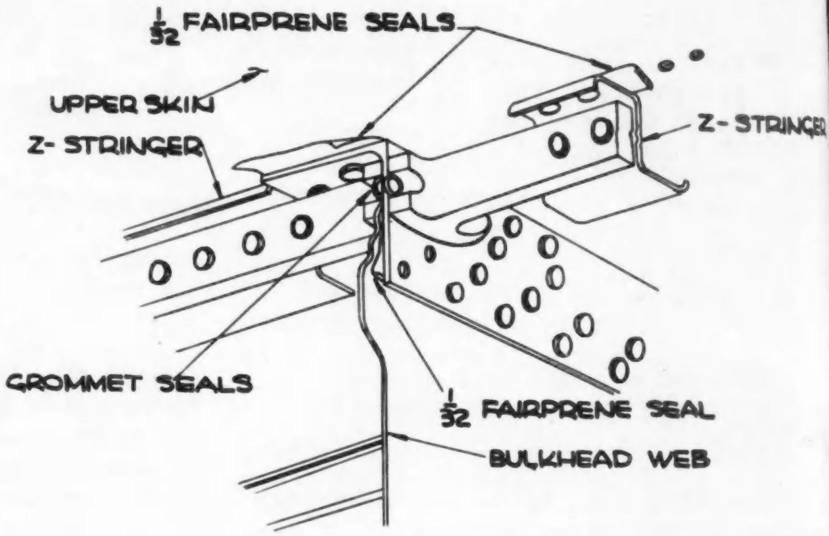
Their feelings don't seem to be justified, because of the B-314 and the PBY service experience. It should be pointed out, though, that the hydrostabilizers of the B-314 were designed for high water loads during take-offs and landings, so that the joints were subjected to high stress only during this portion of the operation and only to a negligible amount during normal flight operation. Consequently, the total working of this structure for a given life would probably be less than that encountered in a flexible wing integral tank structure.

Meanwhile, work has been proceeding on the development of tanks that obtain fuel tightness by means of adhesive sealants over all joints and rivets. The most widely used sealants are: a zinc chromate compound designated RL-3700, thiokol compounds EC-801 and LAC putty No. 5, and an ammonia-cured thiokol compound called Stoner-Mudge. Experience has shown that thiokol compounds are the most adherent and provide the best sealing, with the exception of ammonia-cured LP-2. Some trouble has been experienced, due to the lack of a satisfactory stripping compound to remove EC-801, as required during repairs. Zinc chromate does not stick as well, but it is easy to remove and make repairs, and it does not require aging, as do EC-801 and LAC putty No. 5.

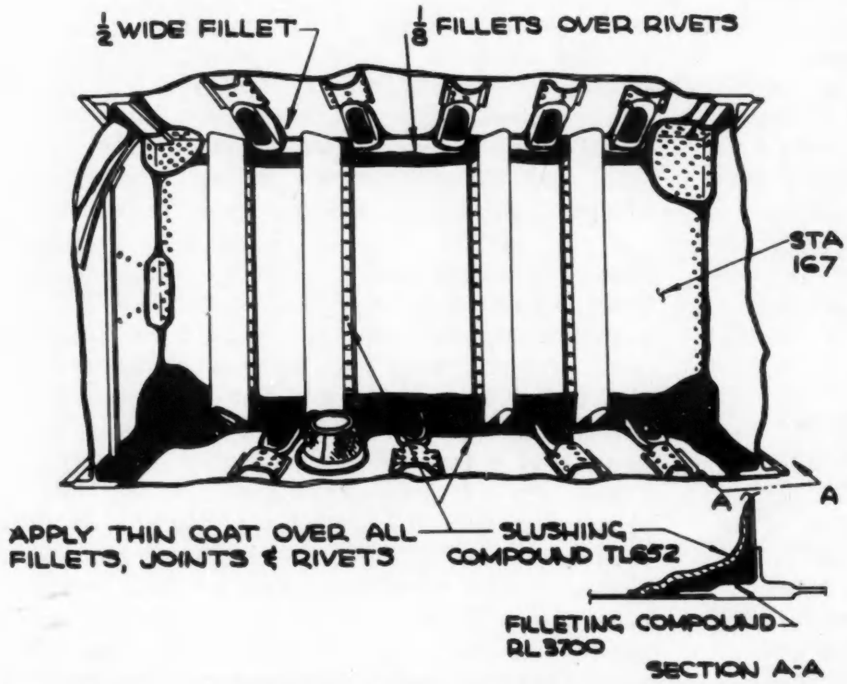
Ammonia-cured thiokol compounds require little maintenance when new, but with aging they shrink away from the walls of the tank, leading to cracks and leaks.

All these materials show a rapid increase in maintenance with age, so that replacement or rejuvenation of the sealant appears desirable, but the cost of replacing the adhesive sealant of integral tanks is almost prohibitive. In Pan American experience, nowhere today is there an adhesive sealant with a satisfactory service record.

The application of adhesive sealant to DC-4 integral tanks is shown in Fig. 4. A fillet of zinc chromate is applied to the inside surface and corners, faying surfaces, and structural joints, as shown in black. A waterproof slushing compound is applied in a thin coat over the zinc chromate fillets, and over the rivets and structural joints. Fig. 5 shows a



■ Fig. 3 (above) Convair 240 integral tank Z-stringer splice bolted end fittings



■ Fig. 4 (left) - Douglas C-54 integral tank sealed with fillet and slushing compound

fuel tank leak chart for one aircraft for a 4-month maintenance period. Each successive leak and repair is noted by consecutive numbers. Many repairs were required at some locations, indicating that the present repair procedure and materials are not adequate.

The tanks on Pan American L-49's, were sealed by first cleaning the tank with ethyl acetate, then applying Lockheed No. 2 putty along all joining surfaces, finally coating the putty with Goodyear No. 30 cement to give a water-resistant coating. The first months of operation were practically free from gasoline leaks, but then trouble was experienced with blisters and bubbles, followed by sealant loosening and cracking, due to the loss of adhesive properties between the sealant and the metal.

Blisters may lift the sealant materials completely from the metal structure, causing the sealant to thin out in wall thickness and thus become a potential source of leakage. They may grow to several inches in diameter. According to Lockheed they are due to the cleaning processes used in the tank cleaning prior to the application of the sealant.

Bubbles, on the other hand, are a delamination of the top coat material caused by heat, such as the sun's rays on the wing surface when tanks are low in fuel. The apparent cause is vaporization of the fuel absorbed by the sealing compound. Bubbles form just underneath the surface of the Goodyear No. 30 cement coat, leaving most of the sealant thickness intact to seal the tank. Generally, bubbles are under $\frac{1}{2}$-in. diameter, although when groups of them grow together, a break may occur and leakage develop. Individual bubbles will not cause leaks. This trouble should now be eliminated, as Goodyear No. 30 (the only material with which bubbles occurred) has been replaced by Nos. 5 and 5F putty.

Latest design trends are indicated by the sealant methods used in the DC-6 integral tanks. A minimum of three rows of sleeve-type rivets is incorporated wherever possible. These rivets have a separate

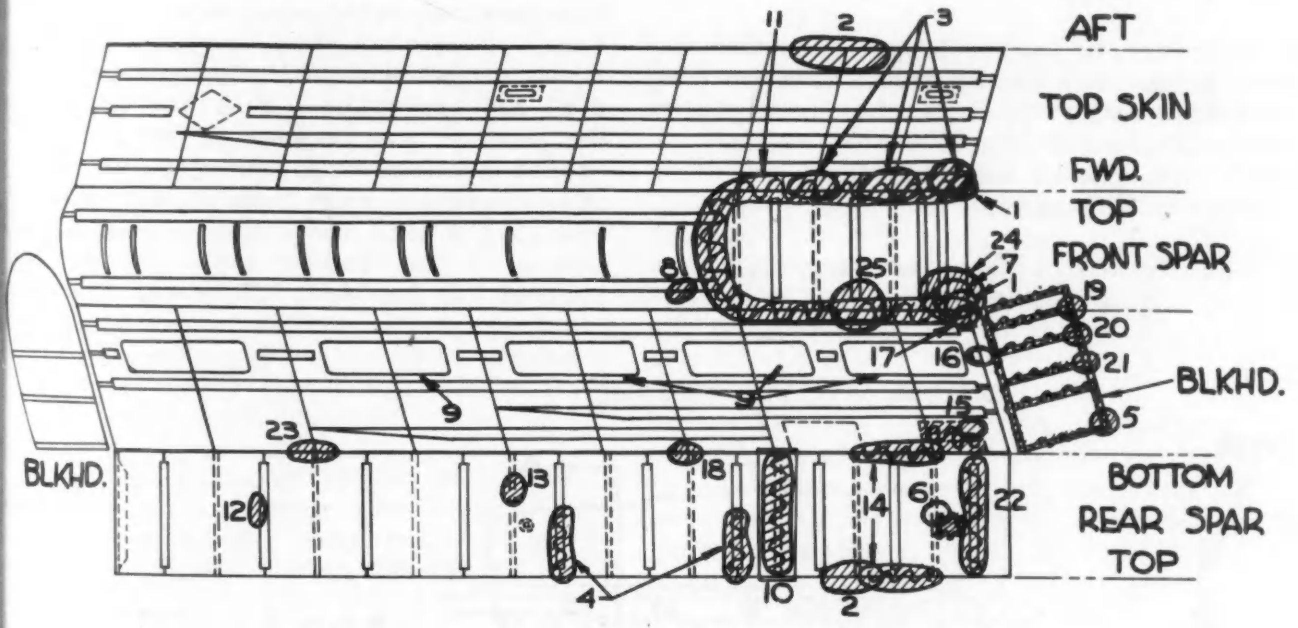


Fig. 5 - Fuel tank leak and repair chart for C-54E airplane; RL-3700 and TL-652 sealant; age of sealant: 6400 hr; period of data: April through July, 1947 - tank completely slushed during this period

O - Leak indication located (external)

□ - Repair location (external)

- Internal reseal with RL-3700

- Internal reseal with slushing compound

outer sleeve, which substantially seals up the rivet hole when the rivet is driven. In addition, as shown in Fig. 6, a nylon thread is incorporated in the spar web to the spar cap riveted seams, and sealing compound is applied to the seams. Although these procedures provide sealant in most of the tank seams and partially seal the tank, the primary sealant is EC-801 applied to all seams inside the tank. Spar stiffeners are placed on the outside of the spars, which simplifies the problem of making a sealed tank. The limited service experience to date with these tanks has been good.

RUBBER CELLS

There are three basic types of nonmetallic cells:

1. Bullet sealing: The tank is of sandwich construction consisting of Buna N inner liner, nylon barrier, and laminations of sealing gum and fabric. Weight varies from 1.8 to 3 psf, depending on the number of outer stiffening fabric layers. They were developed to a high degree in World War II.

2. Nonmetallic tanks: Essentially the same as bullet-sealing tanks with sealing layers and additional fabric for rigidity and puncture resistance. They weigh about the same as aluminum tanks of equivalent capacity.

3. Bladder cells: These are very light weight, nonmetallic tanks. One type contains Buna N inner liner, a nylon barrier, and a fabric outer cover. The other type is made of two layers of nylon fabric impregnated with nylon. Weight varies from 0.3 to 0.075 psf.

All the major rubber companies are thought to have bladder cells approved by the Armed Services and the CAA.

In the Boeing 377 Stratocruiser, 35 nylon bladder fuel cells are connected together by interconnect fittings to form five separate fuel tanks. Thirty-two cells are located in cavities formed by the front and rear spars and the wing ribs. The three remaining ones form the center tank. These cells are attached to their compartment liners by glove fasteners. Cell leaks may cause gasoline to flow into the surrounding structure, whereas integral tank leaks are generally directly to the outside. According to the manufacturer, this can be prevented from becoming a hazard by sealing all fuel tank compartments to provide an overrich mixture if a leak occurs. Drain lines to the lower wing surface provide drainage of each compartment.

Pan American experience with bladder tanks is limited, but has been satisfactory, except for cell plumbing leaks in C-87 ATC operation. They are being used for part of the tankage in the C-54. United Air Lines reports that in their DC-6 operation they have encountered difficulty with the support attachment points of the bladder tanks provided for part of the tankage. This was apparent-

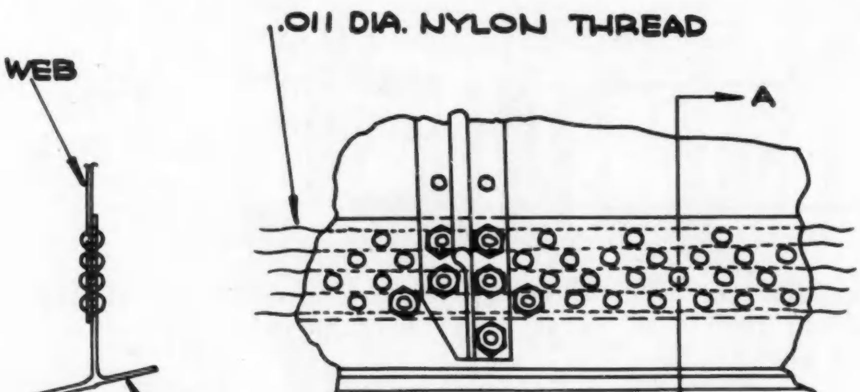


Fig. 6 - DC-6 integral tank - typical installation of seal thread at spar web to cap joint

ly a detail that can be corrected. They have recently had five cases of bladder cell leaking, whereas so far they have experienced no difficulty with their DC-6 integral tanks (their planes have 1000 hr of service so far).

At present there is insufficient experience to say what types of troubles we may encounter with the new, lighter bladder fuel cells planned for such planes as the B-377. Boeing reports that gasoline tends to remove the original plasticizer, but fuel acts as a plasticizer in service. This characteristic makes it necessary to coat the inner surface of the cells with oil when the cells are out of service for at least ten days.

(Complete paper on which this article is based is available from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Summary of Discussion

Integral fuel tanks seem to be finding greatest favor for transport aircraft, despite some sad experiences with early designs, although bladder cells are useful in spaces unsuitable for integral tanks.

Bladder cells are less satisfactory for holding the major portion of fuel, it was pointed out, because they weigh more and have less capacity than integral tanks. Several other disadvantages were enumerated, such as:

1. A greater fire and explosion hazard resulting from cells leaking into structural compartments.

2. Leak detection is difficult because of plumbing requirements. Insertion and removal of cells may be necessary several times with complicated systems before complete sealing is obtained.

3. Structural failures in the tank area can reach serious proportions before they are discovered. Such failures are revealed immediately with integral fuel tanks.

It was explained that little leakage has been encountered with bladder cells so far, but that experience has not yet been sufficient to pass final judgment on them. One discusser felt that if bladder cells were designed into the wing from the beginning, it might be possible to make them weigh about the same as an integral tank.

Recent experience with integral tanks was reported to be most gratifying, due to the development of improved sealants, better manufacturing and maintenance techniques, and a recognition of certain design limitations.

Important has been the realization that sealing material between faying surfaces reduces the structural strength materially, especially when a structure is subjected to high reversal loads. It follows that to design a tank with caulked joints it is necessary to use heavier structure and to make sure that its members are attached so as to eliminate all possible motion, which calls for extra rivets.

It was pointed out further that since sealing materials used in caulked tanks depend on the swelling of the caulking material to prevent leakage, if the tanks are allowed to dry out, they will usually leak when they are refilled until the caulking material has had a chance to swell.

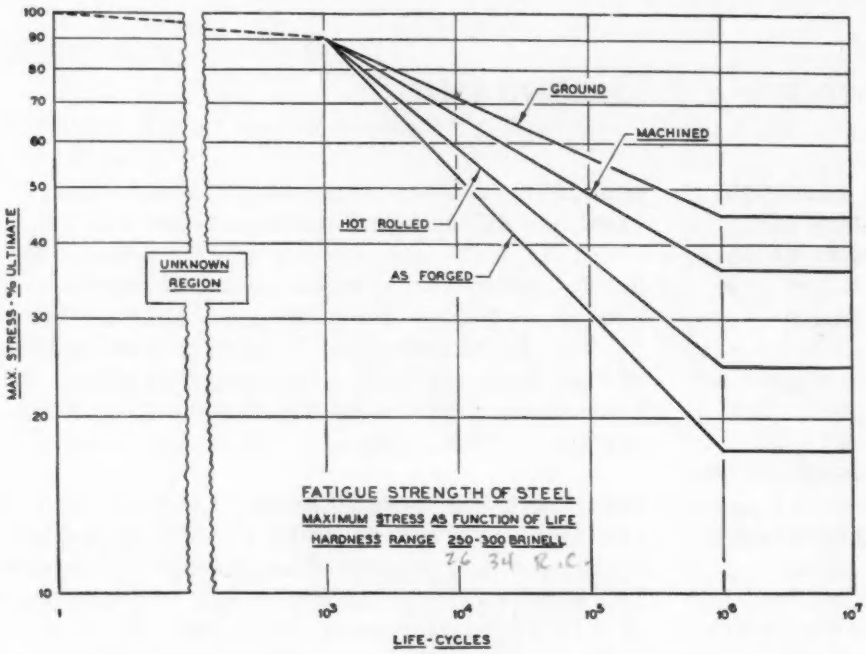
Such troubles are eliminated, it was said, when membrane tanks are used. It was pointed out that the blisters and bubbles discussed by the authors are now fairly well understood and steps are being taken to correct them.

BASED ON A PAPER* BY

Walter E. Jominy

Staff Engineer, Chrysler Corp.

EXPLAIN



■ Fig. 1 - Fatigue strength of steel is a function of surface condition

BECAUSE we've narrowed down criteria for steel selection from a multitude of assorted properties to a mechanical performance basis, hardenability demands attention as a steel-buying tool. It takes into account essential physical properties such as tensile strength, endurance limit, and impact resistance.

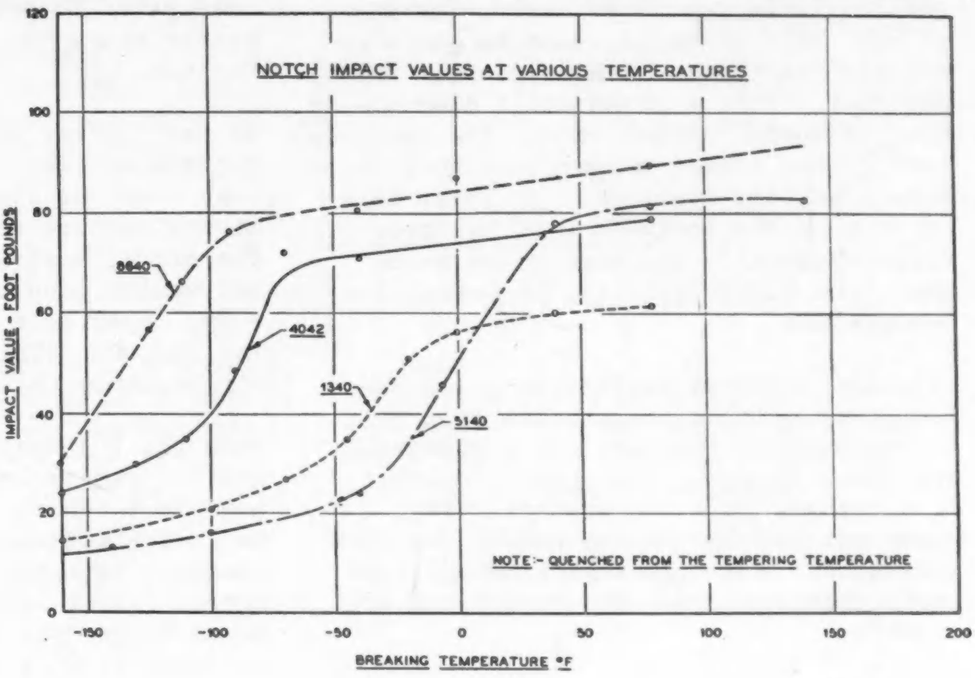
Important criteria-reducer is the knowledge that common SAE alloy steels don't differ much in engineering properties if they're properly heat-treated. Biggest difference between them is hardenability. Producing a tempered martensitic structure in highly-stressed locations in the part by heat-treatment gives us the desired hardenability.

Ability of heat-treatment to impart the same tensile properties to one standard alloy steel as to any other (in the 200 to 500 Brinell hardness range) greatly simplifies the steel-choosing job. It means that the most expensive and least expensive alloys have the same physical properties - when properly heat-treated. Joker here is the "when properly heat-treated" phrase. Large-section parts cannot be heat-treated properly unless the steel has sufficient hardenability; and you can't get high hardenability, if you need it, in lowest-price steel alloys.

Since tensile strengths of common SAE alloy steels are the same if properly

STEEL HARDENABILITY

• WHAT IT IS
• HOW IT'S USED



■ Fig. 2 - Notch impact values of several steels plotted against breaking temperature

heat-treated, endurance limits of these steels should be about the same since endurance is related to tensile strength. Endurance limit is important because automotive steel's service performance is predicated on its ability to withstand fatigue loads. Rule-of-thumb approximation of endurance strength is 50% of tensile strength. Endurance limits for machined and forged surfaces are much lower.

* Paper "A Present-Day Approach to the Choice and Application of Automotive Steels," was presented at SAE Annual Meeting, Detroit, Jan. 12, 1948.

Fig. 1 shows the S-N curves for steel in the 250-300 Brinell hardness range with surface finishes from ground to as-forged conditions. It shows endurance limit to vary from 45% of ultimate strength for a ground surface to 18% for steel with an as-forged surface. But this surface-finish effect diminishes with a reduction in the number of cycles to failure.

If hardness is lower than the 250-300 Brinell range in Fig. 1, percentage of tensile strength to give the endurance limit for the as-forged surface increases; as hardness increases, the percentage value decreases. The as-forged surface endurance limit at 175 Brinell is

about 25% of ultimate strength compared to 10% at the 500 Brinell range. Endurance limit for a ground surface changes in the same direction, but not nearly as much. For example, endurance limit for the 175 Brinell range is 50% of tensile strength, but at 500 Brinell it's 40%.

Impact resistance is another steel property, given considerable attention in recent years, which the hardenability theory embodies.

An important factor influencing impact-test results is the temperature below which brittle fractures occur. The lower the temperature required to get a brittle fracture, the more ductile the material. With a given steel composition, tempered martensite is the structure giving lowest temperature to produce a brittle fracture. At least this is true in the Rockwell C 50 hardness range. Generally the higher the hardness, the higher the brittle transition temperature.

Although tempered martensite gives the lowest transition temperature from brittle to ductile fracture for a given composition, changing the steel's composition changes this temperature. Fig. 2 shows notched-bar impact values for various steels when quenched from the tempering temperature. The variations are sizable.

ROLE OF MARTENSITE

Key to understanding steel's behavior was the discovery that tempered martensite gives best mechanical properties that heat-treatment can develop, particularly best impact resistance, percent elongation and reduction of area with a given tensile strength. It also gives best fatigue resistance for hardness below Rockwell C 50. Figs. 3 and 4 illustrate this superiority of tempered martensite. Alloy additions modify these properties very little, except impact-notch resistance. This is a corollary to the observation that proper heat-treatment develops the same physical properties in any alloy steel. In other words, the heat-treatment that produces martensite during quenching is the right one.

We want to get martensite during the quench when hardening steel because it gives top performance attainable with a given application for most applications. Getting 100% martensite is desirable, but not much is lost with a little less, say only 90% and perhaps some bainite.

And you don't need a martensitic structure completely through the core. Here's why: In most applications bending or torsional forces impose the highest load at the surface; these stresses decrease to zero at the center. If we get the right strength at the surface, there's some point between the surface and the center where the stresses become unimportant.

At one-quarter of the distance between surface and center, the stress is 25% less than the surface stress. So poorer microstructure can be tolerated here. Therefore, a structure with about 90% martensite need be developed only in the outer layer to a distance of one-quarter the radius. (This doesn't apply for parts mainly in tension, such as bolts.)

Both the 90% martensite and the one-quarter radius layer are compromise figures that can be varied with circumstances. The 90% martensite corresponds to hardness acceptable in good practice. It agrees fairly well with the hardness given in the SAE Handbook for limiting hardness with carbon content in the end-quench test.

Examining the microstructure for percent of martensite is a slow, laborious process; but it's easy and generally adequate to measure hardness that corresponds to the microstructures. Hardness corresponding to percent martensite compared to carbon content is given in Fig. 5. For a given carbon content, these curves tell the hardness equivalent to 90% martensite - which quenching produces to one-quarter the depth of the radius.

While composition hardly affects engineering properties of alloy steels (provided we have tempered martensite), it does influence hardenability. For this reason choice of steel for any part depends largely on its hardenability.

For example, highest carbon content gives most hardenability per unit cost for any given series of medium alloy steel. Fig. 6 shows relationship between carbon content and hardenability for the 8600 series. Criterion of hardenability in each case is the size which will harden with a minimum of 90% martensite to a depth of one-quarter the radius. It shows that you can increase the hardenability from a 0.4-in. to a 1.9-in. round just by stepping up carbon from 0.17% to 0.53%. There is no difference in cost between these steels. Of course fabrication difficulties increase with higher carbon steels.

Boegehold[1] developed a method for choosing a steel of correct hardenability by sectioning the part where good microstructure is needed and by measuring hardness at that point. Here is how you do it:

[1]See SAE Transactions, Vol. 52, October, 1944, pp. 472-485: "Selection of Automotive Steel on the Basis of Hardenability," by A.L. Boegehold.

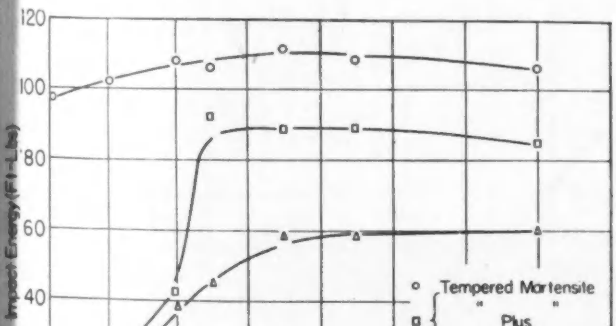


Fig. 3 - Tempered martensite structure gives best impact resistance, these tests at a tensile strength of about 125,000 psi show

From the same steel from which the part is made, machine a standard end-cooled bar and determine the hardenability of the steel. Compare the hardness of the sectioned part with that on the end-cooled bar. From this you can determine cooling rate at the critical sections. Now you can choose the steel that will harden satisfactorily with this cooling rate.

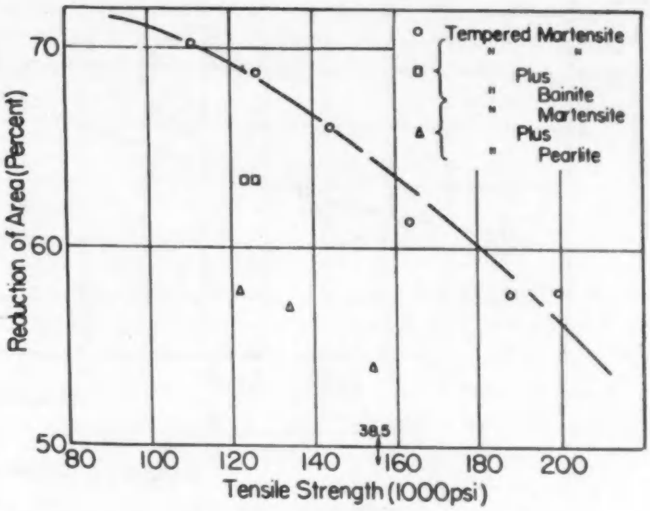


Fig. 4 - Tempered martensite is also the best-performing steel structure from a reduction-of-area standpoint

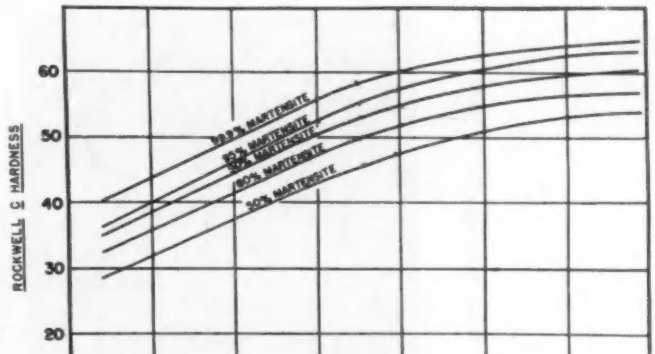
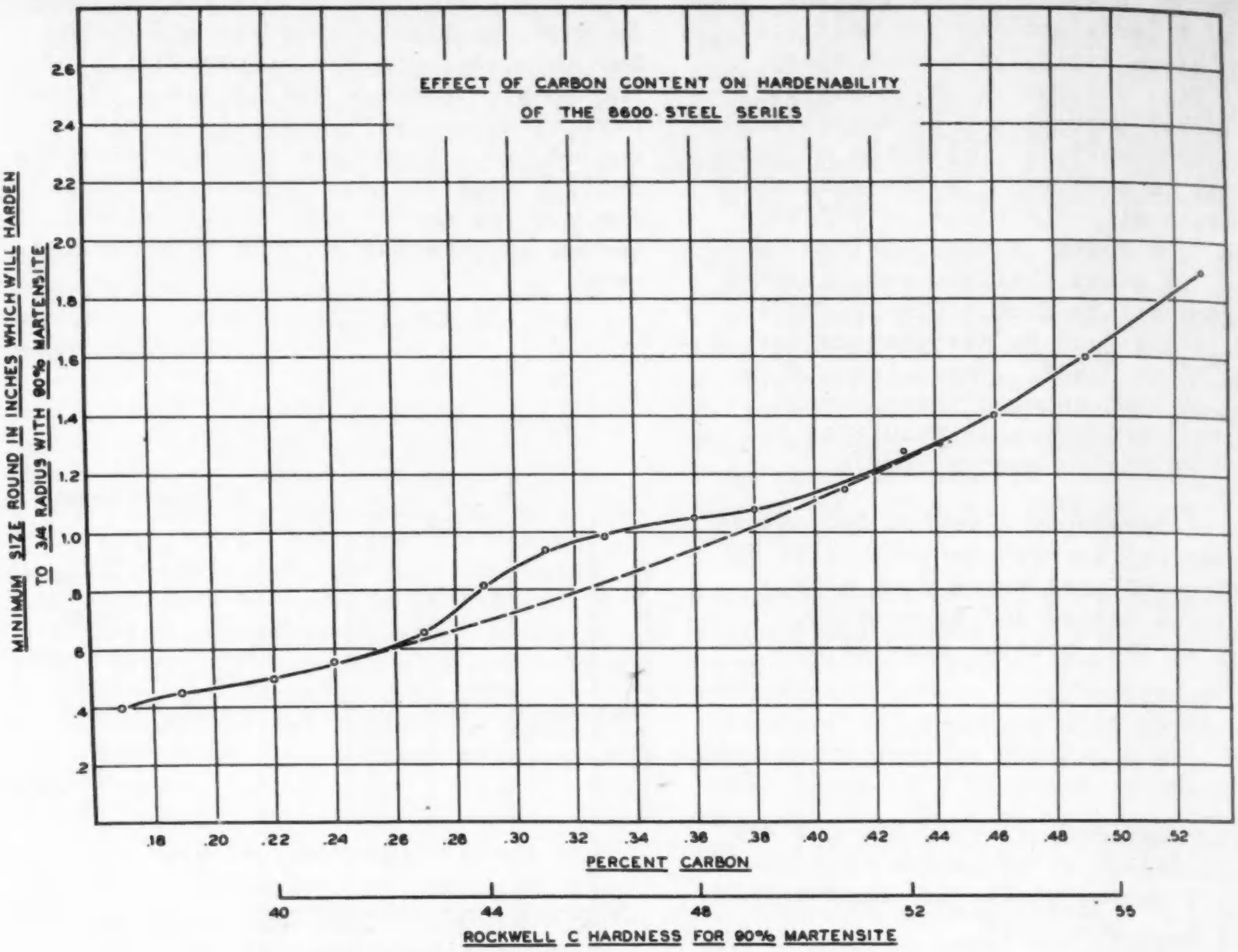
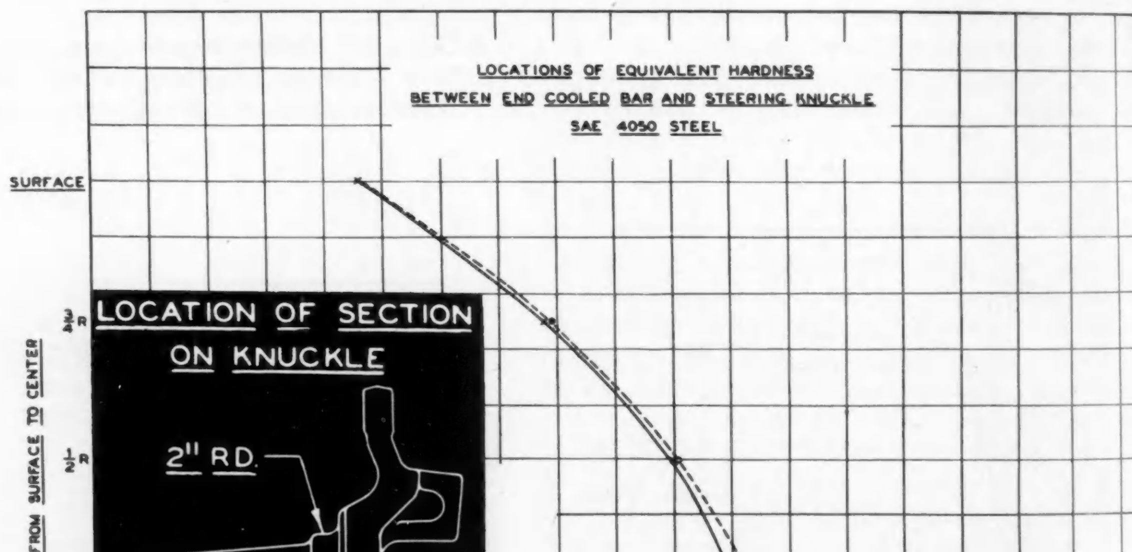


Fig. 5 - This chart shows the relationships between carbon content, hardness, and percentage of martensite


EFFECT OF CARBON CONTENT ON HARDENABILITY
OF THE 8600 STEEL SERIES
MINIMUM SIZE ROUND IN INCHES WHICH WILL HARDEN
TO 3/4 RADIUS WITH 90% MARTENSITE
2.6
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
.8
.6
.4
.2
.18 .20 .22 .24 .26 .28 .30 .32 .34 .36 .38 .40 .42 .44 .46 .48 .50 .52
PERCENT CARBON
40 44 48 52 55
ROCKWELL C HARDNESS FOR 90% MARTENSITE


LOCATIONS OF EQUIVALENT HARDNESS
BETWEEN END COOLED BAR AND STEERING KNUCKLE
SAE 4050 STEEL
SURFACE
3/4 R
1/2 R
CENTER
DEPTH FROM SURFACE TO CENTER
LOCATION OF SECTION
ON KNUCKLE
2" RD.
STEERING KNUCKLE
2 INCH ROUND
2 4 6 8 10 12 14 16 18
DISTANCE FROM END - SIXTEENTHS

Let's go through a typical shop problem on a part such as the irregularly shaped car steering knuckle, shown in Fig. 7. It is hardened by quenching in oil from a suitable austenizing temperature.

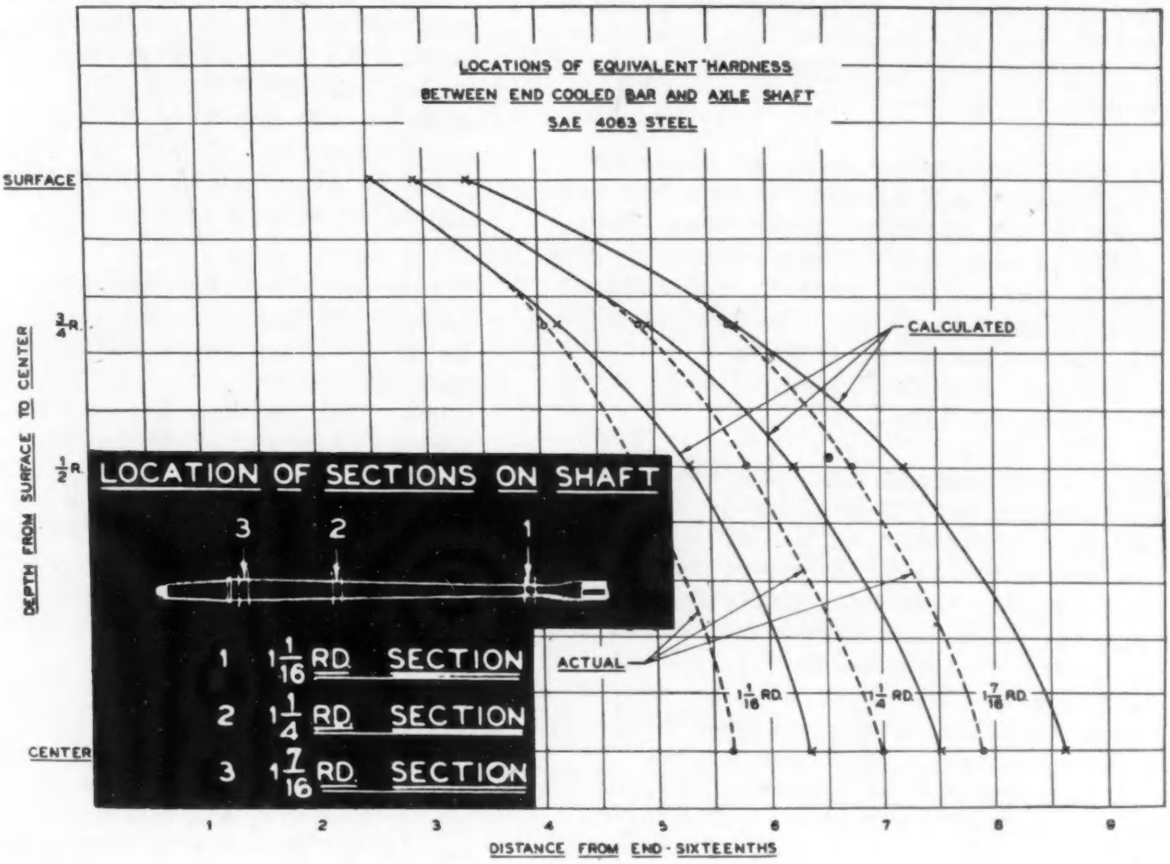
Fig. 6 - Composition influences steel's hardenability. This curve depicts the variation in hardness with carbon content for the 8600 steel series

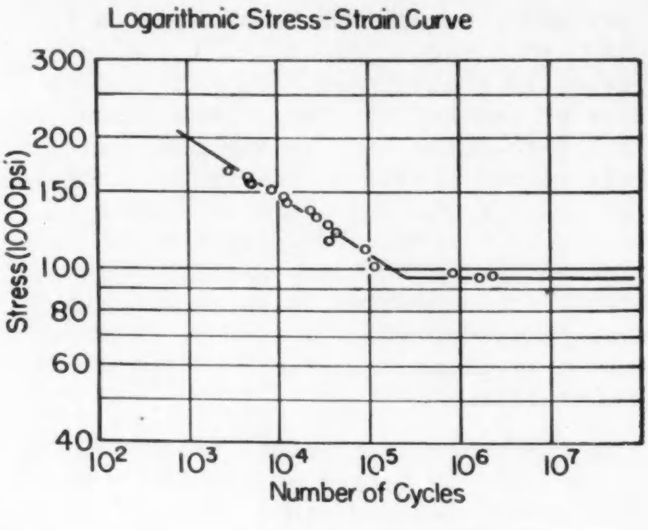
Fig. 7 - The solid curve represents cooling rate for a 2-in. round test bar, made from the same steel as the steering knuckle shown, which was quenched in still oil. The dotted line compares points of identical hardness on the test bar with that in the critical section of the part (just to the left of the arrow)

First, we want to know locations on this part most susceptible to service failure. At these points we want a microstructure giving best physical properties we can get in steel. Weak point of this particular part is the one just left of the arrow, at the radius between the 2-in. round and the tapered sections. To determine cooling rate at this location, we follow the part through production hardening, then section it through the critical location. Next we make a hardness traverse from surface to center of the section.

From the same bar of steel from which the knuckle is forged we machine a standard end-cooled hardenability bar, hard-

Fig. 8 - A rear axle drive with cooling rates at its three critical sections. The curves reveal the production cooling rates to be faster than oil quenching of equivalent sections





Logarithmic S-N Curve

■Fig. 9 - Endurance life curve for steel with a hardness of Rockwell C 39

ened to conform with the standard hardenability test. We now compare the hardness on the end-cooled bar with that obtained on the cross-section of the piece. We assume that points of equal hardness will cool at the same rate. Having found the cooling rates for all points on the end-cooled bar, we can figure the cooling rate at various points in the section.

In this particular case two steering knuckles were followed through the processing and hardness traverses. The dotted curve in Fig. 7 was gotten by comparing points of identical hardness on the bar and section. The cooling rate is quite similar to that in a 2-in. round bar quenched in still oil, shown by the solid line.

This curve reveals that any steel that will harden to 90% martensite at 10/16 in. on the end-cooled bar will produce the 90% martensite at one-half the radius of the steering knuckle's critical section. Also any steel that will harden to 90% martensite at the 8/16-in. point on the test part will harden to 90% martensite at the one-quarter point on the section. This then is what we use in going to the H-band specifications to find the right steel for the job.

Let's study a part with three critical sections. It's the rear-axle drive in Fig. 8. The dotted curves show actual cooling rates in production are faster than oil quenching equal size sections. These curves show that steel hardening to the three-quarter radius point in the 1 7/16-in. round will harden completely through the 1 1/16-in. round section. Therefore, it's impossible to maintain a soft core in the 1 1/16-in. section and still harden the 1 7/16-in. section to the three-quarter radius.

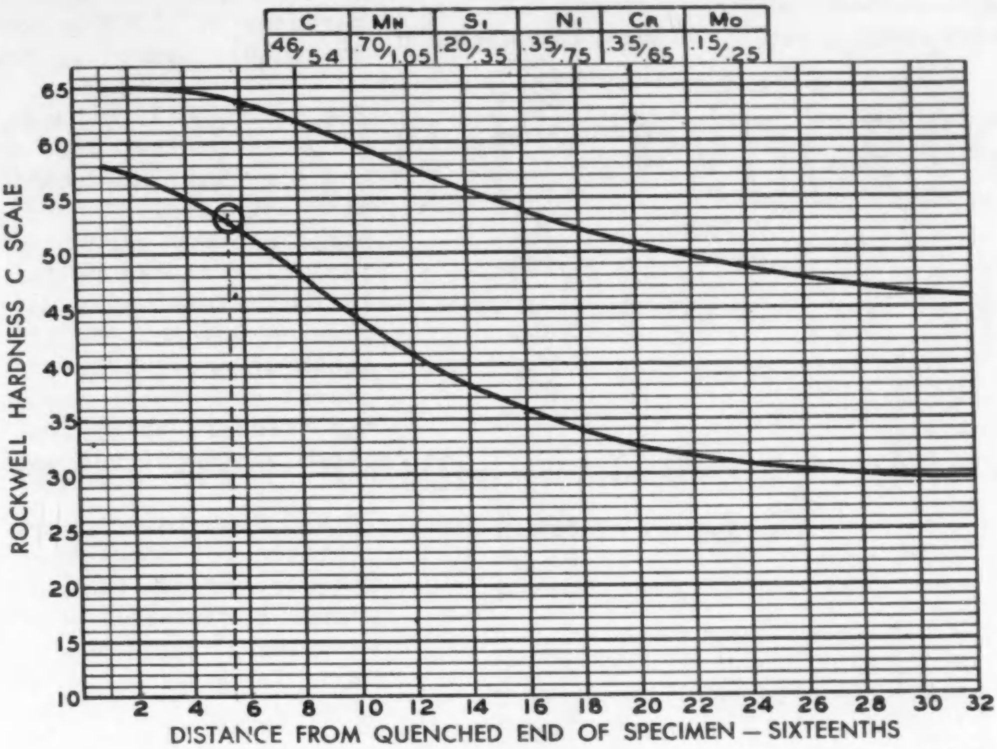
Production experience shows that for this shaft, steels on the low side of hardenability distort less than those on the high side. Thus for best results we prefer steels having 90% martensite at not less than 11/32 and not more than 13/32 in. Since such close specifications are impossible, we must accept steels for this shaft with deeper hardenability and more distortion than we prefer.

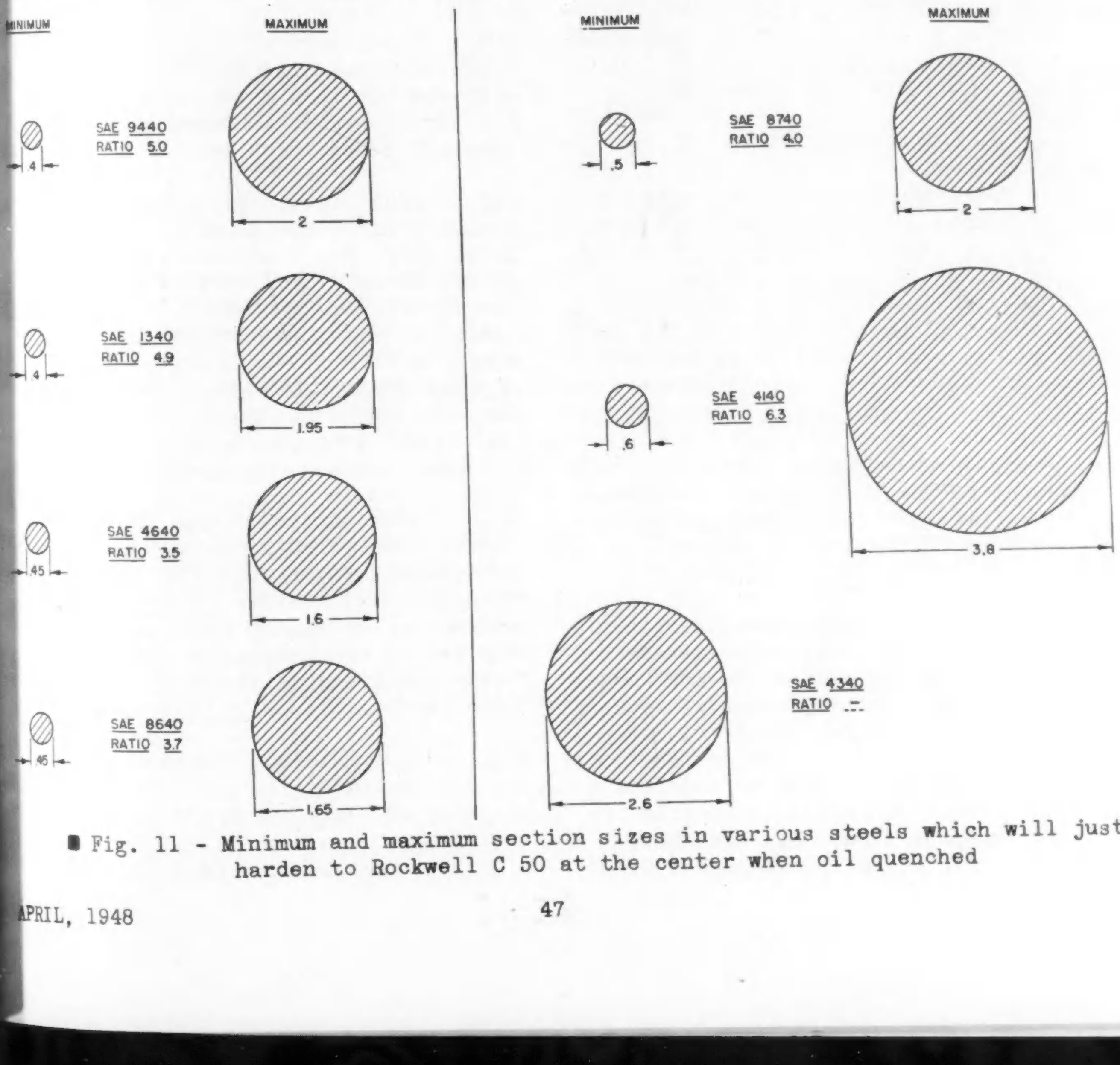
Suppose we take this same part and see how the designer would choose steel for it as a new part or substitutes steel for it if already in production. Assume maximum stress in the section of location 3 is 150,000 psi from combined torsional and bending loads. This maximum stress, the designer tells us, occurs only occasionally and the usual fatigue load is 25,000 psi. To minimize distortion, we will quench in oil.

Choice of steel and heat-treatment for such a shaft hinges on its going through as many stress reversals at 150,000 psi without failure as it's apt to in service. Fig. 9 is a typical endurance life curve for steel of Rockwell C 39 hardness, if it's martensitic. Going back to Fig. 1, we see that the part, if finished ground, will withstand about 10,000 applications of a 150,000-psi load. In the as-forged condition it will withstand only about 3000 such load applications.

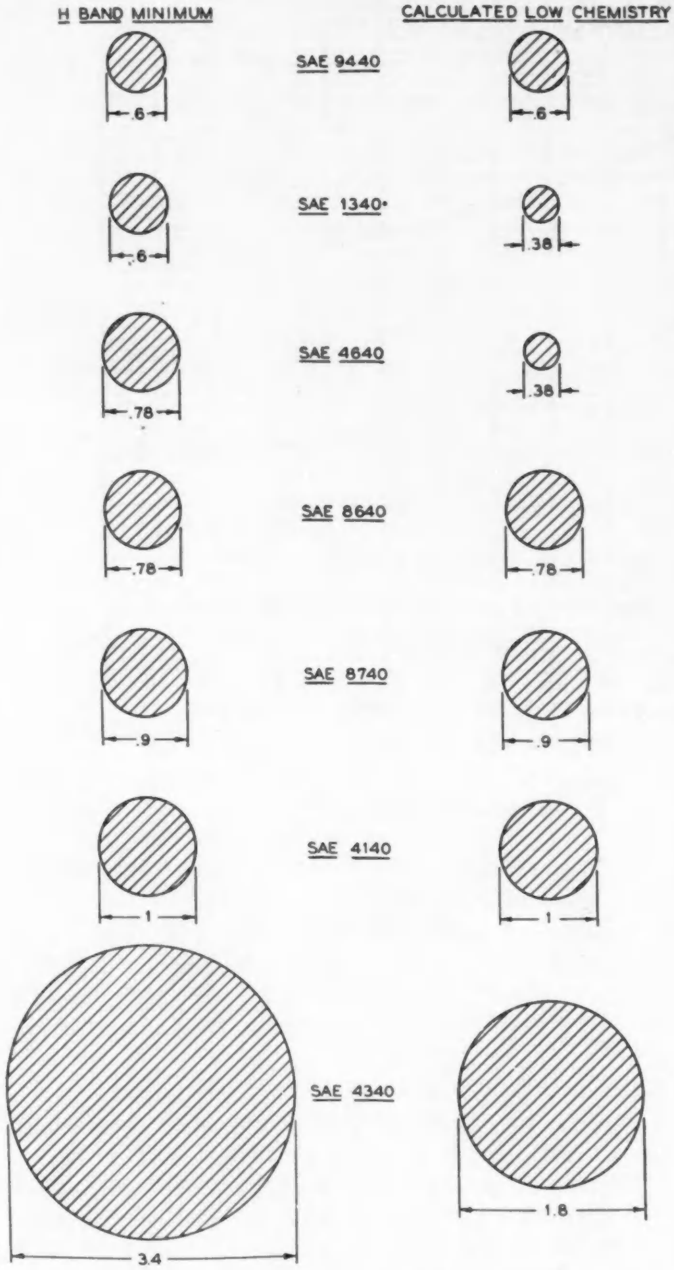
Suppose the experienced engineer says that 3000 load applications are more than the part will get in service and let's accept a minimum hardness of Rockwell C 39 as satisfactory. Since the

■ Fig. 10 - H-band for SAE 8650 H steel. It satisfies needs of the rear - axle drive in Fig. 8 because it has a minimum hardenability of 11/32 in., with a Rockwell C 53 hardness





■ Fig. 11 - Minimum and maximum section sizes in various steels which will just harden to Rockwell C 50 at the center when oil quenched



■ Fig. 12 - Bar sizes which will just harden to Rockwell C 50 at three-quarter radius when oil quenched

stress at the surface is 150,000 psi and only 102,500 psi at the three-quarter radius distance, we'll need the 90% martensite for only the surface-to-three-quarter-radius layer on the 1 7/16-in. round section. Fig. 8 tells us that steel chosen should have a minimum hardenability of 11/32 in. It's also desirable to select a steel with a carbon content giving at least a Rockwell C 50 hardness with 90% martensite to get a reasonable tempering temperature.

The hardenability bands in the SAE-AISI manual show that 8650 H steel, Fig. 10, has a minimum hardenability of 11/32 in. with a Rockwell C 53 hardness. It also has a minimum carbon content of 0.44%, including allowance for permissible variation. This tells us that 8650 H steel gives a 90% martensitic structure to a depth of one-quarter radius with its lowest acceptable hardenability under the standard specification. This steel is suitable for the part.

If SAE 8650 is not available or not acceptable for other-than-hardenability reasons, we can find other steels giving proper hardenability properties. For example, if only one-tenth of the radius needs to be hardened, SAE 5145 could be used.

H-BAND CHARACTERISTICS

Prospective users of the hardenability-band specifications for standard steels will want to know more about their range and advantages as compared with the chemical composition specifications.

Rather wide limits have been adopted as standard for H-bands to which steel can be bought. Fig. 11 gives some notion of the spread. It illustrates the maximum and minimum bar sizes that will harden to the center to Rockwell C 50 when steel is on both the lowest and highest acceptable side of hardenability, according to SAE-AISI standards. The higher the ratio between maximum and minimum diameter, the greater the spread.

The illustration shows that if a 0.5-in. round steel bar is to be hardened through the center, SAE 8740 or a higher hardenability steel will be needed. But since It's seldom necessary to harden steel to the center, we can use steels for larger minimum bar sizes than those in Fig. 11.

Fig. 12 shows minimum bar sizes for these same steels if we have to harden to depth of only one-quarter radius to a

Concluded on page 58

DIESEL DESIGN PROGRESS SPEARHEADS

Penetration of Prime Mover Fields

BASED ON PAPER* BY

Saul Belilove

Engine Division, Enterprise Engine & Foundry Co.

RESPONSIBLE for growing diesel popularity is its inherently high efficiency compared with other powerplants and improvements in getting more power from the same engine size. Diesel developments being pushed in both these directions are recruiting an increasing following among prime mover users.

Biggest diesel advantage lies in the fact that it's the most efficient heat engine known - its fuel consumption is lowest. For this reason it has practically obsoleted the steam engine and is even moving into the steam turbine area. The diesel uses cheaper fuels than the gasoline engine. Other reasons why the diesel is outcompeting these two engine types include greater reliability and ruggedness.

Slow speed diesels achieve 40% thermal efficiencies. The diesel's high expansion allows greater efficiencies than the relatively low expansion ratio gasoline engine. But further increase.in diesel expansion ratio is not the way to boost that 40% efficiency.

One way to do it involves a shift in our thinking. Instead of considering the diesel engine as strictly a converter of fuel chemical energy to mechanical energy, let's also visualize it as a producer of heat energy. By considering entire powerplant needs we may learn how to achieve an overall balanced arrangement, using what's now considered waste as an integral by-product of energy output. The engine's thermal efficiency may be only 35 to 40%; but it is possible to use heat rejected to jacket water and exhaust gases for purposes such as space heating and operation of low-pressure boilers for process steam.

Low temperature of this lost heat makes it useless for mechanical work, but it has considerable value in these other applications. If we adopt this viewpoint, we may consider overall use of

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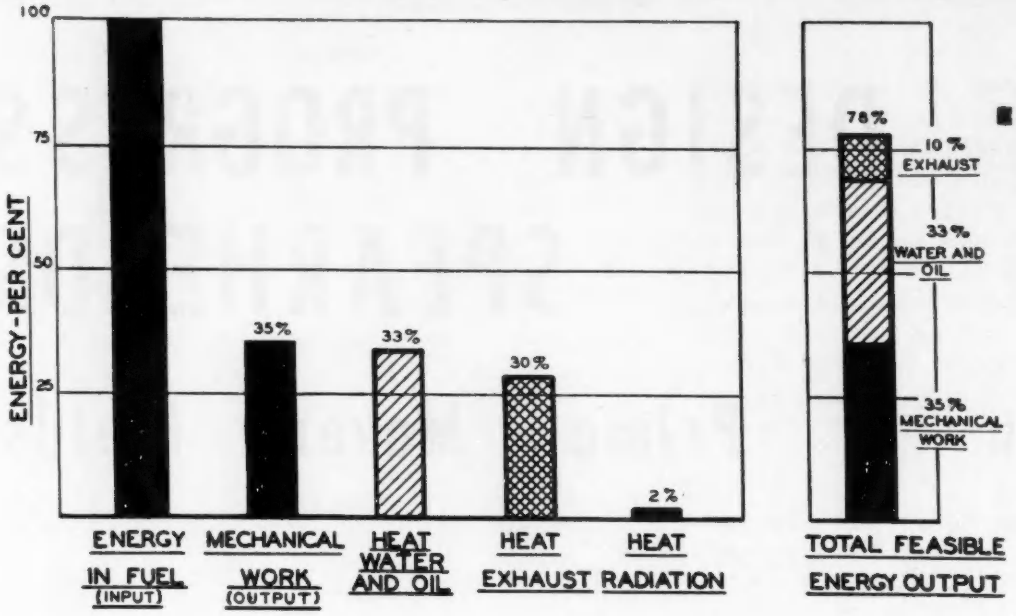
* Paper "Recent and Future Developments in Diesel Engines," was presented at SAE Northern California Section, San Francisco, Oct. 14, 1947.



■ Fig. 1 - How energy in a typical diesel engine could be distributed to realize maximum efficiency

energy within the fuel to be as high as 75% in some plants. See Fig. 1.

A more familiar way of boosting thermal efficiency, used in recent years, is to increase engine loading - greater combustion-gas pressure on the pistons. This raises mechanical efficiency by reducing friction load percentage of the total·load. Supercharged diesels, both two and four-cycle, show somewhat higher thermal efficiencies than their unsupercharged counterparts. Big gain with the exhaust-supercharged four-cycle engine is the significant efficiency improvement at partial loads.

But a more important reason for increasing engine load is to get more power from the same engine size. Greatest recent advances have been made in this aspect of design. Apparently it will have to bear the major development burden in the immediate future.

Greater engine loading or increased average combustion gas pressure demands that we burn more fuel in the cylinder. To burn more fuel we need more oxygen. Supercharging does that by filling the cylinder with greater quantities of fresh air. With supercharging we've boosted average brake mean effective pressures of four-cycle diesels from 80 psi to 120 psi. Today two-cycle engines are operating at 80 psi bmep compared to 60 and 70 psi of the past. There's evidence of increases going even 20 to 25% higher in the near future with four-cycle engines. In fact for intermittent peak-power uses, one manufacturer rates his engine at 150 psi bmep.

TEMPERATURE STYMIES PROGRESS

Designers could get still greater outputs and increased loadings if it were not for a number of limitations, most important of which is heat release. Only a certain amount of heat can be liberated from a cylinder if the piston is to operate satisfactorily. This temperature problem is the current barrier.

At least three recent developments - Heavy-Duty lubricating oils, oil-cooled pistons, and chrome plating - have helped increase the diesel's heat resistance.

Heavy-Duty lubes keep carbon contaminants dispersed in the oil because of their detergent action. Oil-cooled pistons help dissipate the heat radiated and conducted from combustion gases to the piston. Removing the heat from the piston head lowers ring temperature to a safe value so that oil films are maintained and rings operate freely. Chrome-plated piston rings and cylinder liners

also help boost the engine heat tolerance level.

At least two other suggestions to decrease the heat load have been made. The first concerns itself with the fact that intake air temperature plays a big role in determining average gas temperature of the cycle. A simple cooler will decrease intake air temperature sufficiently to raise loadings as much as 20 to 30% above the standard increase with the Buchi system (exhaust-driven supercharger). This could increase the total loading 75% above that of a nonsupercharged engine. An intercooler becomes even more desirable for high-pressure supercharging since heat generated in compressing intake air exceeds that in the Buchi system.

The second recommendation for easing the engine's heat burden consists of cooling the air inside the engine itself by cutting off intake air supply while the piston moves down on its intake stroke. With this system, the engine has a high-pressure exhaust-driven supercharger delivering about 15 psi pressure.

Air enters the engine at this pressure; but since it's cut off before completion of the piston's downward stroke, the remainder of the intake stroke expands this high-pressure air to about 5 or 6 psi (similar to the Buchi system pressure). Air cooling during this expansion achieves the same effect as intercooling. Adoption of this method depends on seriousness of diesel heat limitations; otherwise it's absurd to lower high pressure after going to the trouble of generating it. But it does point up the main road to diesel improvement - generating more heat in the combustion chamber without hurting operating parts.

Because of the diesel's inherently high efficiency and rapid design progress, its use has skyrocketed in this country within the last 10 years. Emphasis on operating economy in the 1930's made for widespread abandonment of the steam engine in favor of the diesel. Improved mechanical design and lubricating oil plus high speeds led to lighter and cheaper engines. These factors have spurred industry's recognition of the diesel.

Let's focus on relative efficiencies of the diesel and other prime mover types, as shown in Fig. 2. The diesel is by far the most efficient. But efficiency is not the sole fuel cost determinant.

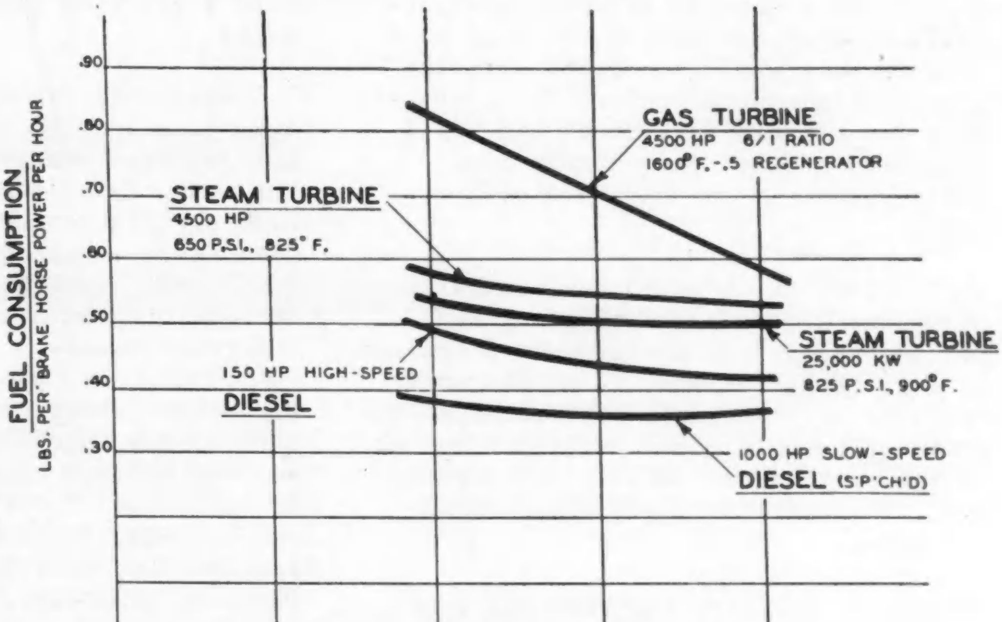


Fig. 2 - Fuel consumption of various types of heat engines

NORTHERN CALIFORNIA (OCTOBER 1947)

FUEL	BASED ON	COST FOR 1 000 000 BTUS
STANDARD DIESEL OIL	8.2¢/GALLON	58¢
BITUMINOUS COAL	$13.00/TON	54¢
BUNKER DIESEL OIL	$2.85/BB	47.5¢
STANDARD FUEL OIL (LIGHT)	$1.80/BB	29¢
NATURAL GAS	30¢/1000 FT³	27¢
BITUMINOUS COAL (NEAR SOURCE)	$5.00/TON	21¢
NATURAL GAS (NEAR SOURCE)	10¢/1000 FT³	9¢

■ Fig. 3 - Fuel costs compared

Cost per unit of fuel is also a salient factor. Fig. 3 shows comparative fuel costs.

In some cases coal is enough cheaper than diesel fuel so that, despite superior diesel efficiency, a coal-burning steam boiler costs less than a diesel running on relatively high-priced oil. But lower per-gallon cost of diesel oil compared to gasoline accounts for many diesels being used instead of gasoline engines. Because of this fuel cost consideration, many are urging lower-grade, low-cost fuel oils. While lower cost fuels adversely affect diesel operation, they can burn these oils at a considerable savings despite slightly higher maintenance costs. This, coupled with the present fuel situation, will stimulate greater concentration on cheaper fuel in the future diesel research.

The current tight petroleum situation shows the disadvantage of popularity. Oil technology can now convert petroleum to suit market demand; it no longer is gasoline or diesel oil or residuals. Instead of being a gasoline by-product as in the past, diesel oil is now a partner. Its cost increases with its importance.

The diesel industry has kept its eyes open to ways of offsetting this mounting fuel price. The combination natural gas-oil engine is a case in point. This engine can burn natural gas on the diesel cycle and attain its full efficiency, using 10% or less oil for pilot ignition. The extremely low cost of natural gas in many parts of the country makes this combination a "natural."

The dual-fuel engine burns fuel oil and natural gas in practically any ratio - from 10% oil and 90% gas to 100% oil. Controls are arranged for automatic changeover from gas to oil operation without disturbing the load. Fig. 4 charts comparable fuel consumptions of the spark ignition, 100% diesel, and dual-fuel engines. Natural gas used in this fashion can halve fuel costs. This adds up to considerable savings, remembering that fuel costs comprise about 75% of direct operating costs.

Proof of the pudding is in the eating - and the inroads being made by the diesel in various fields testify to its growing popularity. Fig. 5 shows how the diesel is used in these areas. Now for a bird's-eye view of diesel progress in each:

1. Railroads: Ten diesel locomotives are now being ordered to every steam job. This demonstrates the complete break of the monopoly held by steam until the 1930's. Big reason is the fact that diesels are available for service over 90% of the time; steam locomotives seemed to spend most of their time in the round-house.

2. Marine: Diesel advantage over steam here is the huge fuel-cost saving as well as release of payload space needed to carry fuel for steam engines. By the end of 1944, total horsepower of installed U.S. Navy diesels exceeded steam for the first time.

3. Automotive: Although handicapped by its greater first cost for trucks, buses

Fig. 4 - These fuel consumption curves demonstrate the merit of combination natural gas-diesel oil engines

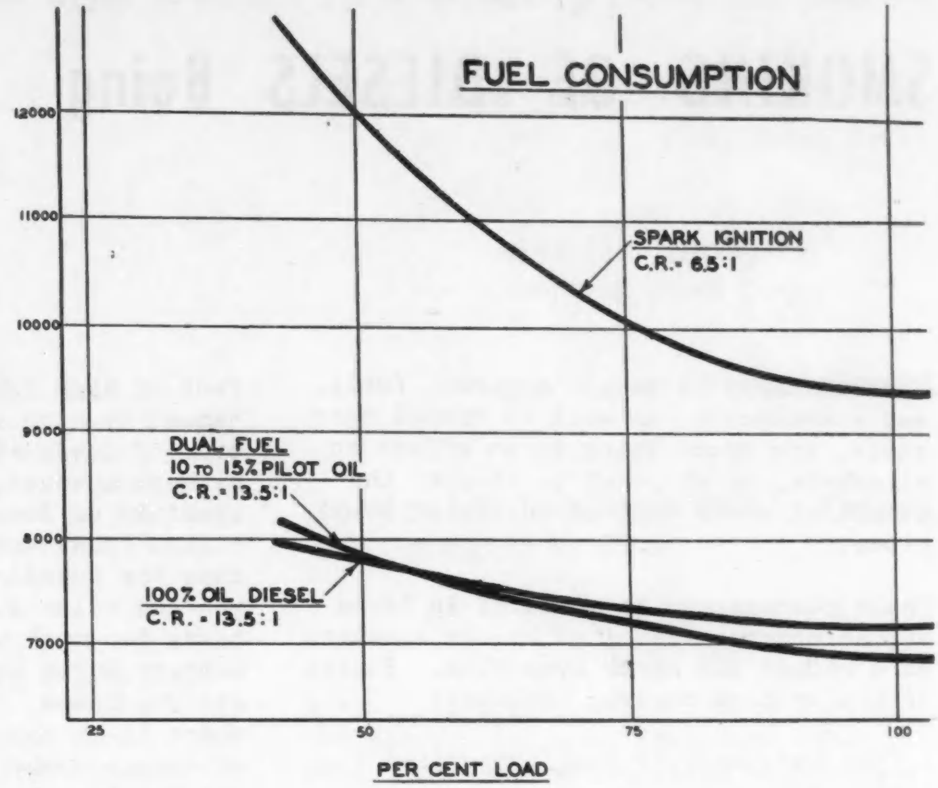
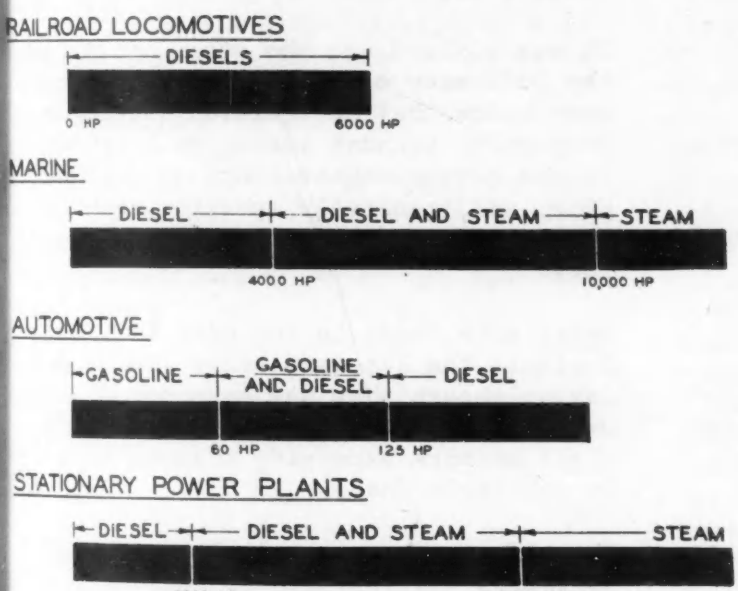


Fig. 5 - How diesel power is being applied in various fields



and tractors, the diesel still nets a fuel cost saving where vehicles operate many thousands of miles per year. Operators now insist on diesel equipment for this reason. Maintenance is still higher than with the gasoline engine, although many will dispute this point. Better diesel torque characteristics improve lugging ability. Because of these advantages, most large trucks and buses placed on the road today are and will continue to be diesel-powered.

4. Stationary: Diesel attraction in the stationary field is its high efficiency, regardless of size. It's compact and can be installed almost as a packaged unit. For isolated communities or for elimination of long transmission lines the diesel is ideal. It also competes with steam in large communities because of electrical transmission problems as well as high overhead charges being borne by many large power companies. It's still far cheaper to transport oil by pipe line or even by rail than to transmit electrical energy over transmission lines.

(Complete paper on which this article is based is available in full from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers)

SMOKING OF DIESELS Being Attacked from

MANUFACTURERS of diesel engines, fuels, and accessories, as well as diesel operators, are cooperating in an effort to eliminate, or at least to reduce, the amount of smoke emitted by diesel engines.

Their success can be measured in terms of the progress reported by the speakers at a recent SAE Smoke Symposium. Phases of the problem covered included:

1. The influence of fuel properties and engine design.

2. A technique for measuring exhaust smoke density.

3. The effect of the smoke curve on engine performance.

4. The part being played by operators in alleviating the problem.

FUEL PROPERTIES

Volatility and ignition quality are the only two properties of fuel that appear to have an appreciable effect on the amount of smoke emitted by an engine.

There was some disagreement, though, as to the relative importance of these properties.

It was stated, on the one hand, that high smoke density is obtained with fuels of high ignition quality or cetane number, due to the short delay period inherent with this type of fuel, which does not allow the fuel spray to develop fully before ignition starts. The heat generated after ignition starts results in intensive evaporation, which hampers further distribution. The adverse effect of high ignition quality is enhanced by high volatility. The penetration of fuels with lower volatility is higher, however, and an appreciable quantity of fuel reaches the fairly hot piston crown and evaporates from there into the boundary air. With fuels of too low volatility, though, evaporation takes too much time, and the resulting mixture burns late in the expansion stroke; hence, for each cetane number, there is an optimum volatility, the lower cetane number requiring a higher volatility.

It was claimed, on the other hand, that the influence of volatility appears to overshadow that of ignition quality in respect to exhaust smoke, at least within the cetane-number range of the engine, and, generally speaking, fuels down to 45 cetane number are satisfactory.

Experience seems to indicate that light fuels in the kerosene range lessen smoking, although they may have poorer economy and, because of distribution costs, might be more expensive than more readily available fuels.

The effect of other volatility characteristics was reported as follows:

1. The 50% point is a fair criterion of exhaust smoke. A satisfactory rule was suggested to be: The higher the sum of the percents of fuel evaporated at a series of fixed temperatures taken at uniform increments between the IBP and the 50% point, the better the smoking characteristics of the fuel.

2. Considerable flexibility is permitted in the selection of the IBP and the EP

BASED ON PAPERS* BY **H. D. Young** Sinclair Refining Co.

Many Angles

J. J. Broeze and C. Stillebroer
Royal Dutch Shell

Martin Berlyn
American Bosch Corp.

Kenneth J. Fleck
Caterpillar Tractor Co.

Jordan P. Jung & Frank L. Coers
Cummins Engine Co., Inc.

A. S. Leonard
Cummins Engine Co., Inc.

of a fuel, provided the distillation curve is pivoted around a 50% point of no more than 475F and the relation between IBP and EP is correct. Thus, if the IBP is high, the EP must be low, and if the IBP is low, the EP can be moderately high.

3. For fuels of a given IBP, exhaust smoke varies directly with tail-end volatility, the lower the tail-end boiling range, the lower the exhaust smoke. For fuels of fixed end point, that having the highest sum of percents distilled and/or the lowest midboiling point will give the best exhaust smoke performance.

ENGINE DESIGN

The precombustion type of engine reacts quite differently to various fuels than does the direct-injection engine, due to the different way in which the fuel is distributed.

In the direct-injection engine, evaporation and air movement must correct the incomplete distribution by the fuel spray. Intensification of one part of the process tends to upset the other part. In the precombustion engine, the injection system has no function in the distribution process, which is performed by evaporation in the precombustion chamber and a blast created by the initial explosion in that chamber. This way is apparently not easily affected by secondary factors, thus reducing the tendency to smoke and sensitiveness toward fuel properties.

Some of the factors involved have been demonstrated by tests of a precombustion chamber of the swirl type. (See Fig. 1.) Lowest smoke density and best consumption were obtained with fuels of 35-40 cetane number, independent of volatility. The walls of the heat-insulated swirl chamber apparently were hot enough for all deposited fuel to evaporate and to burn within a short time. With higher-cetane fuels of low volatility, smoke density and consumption increased because, when ignition started after the shorter delay period, the fuel was more confined in the upper part of the swirl chamber and less vapor was generated. The ensuing pressure rise in the swirl chamber was less and the resulting blast from the chamber, which performed the mixing in the cylinder, was weaker.

Evaporation of fuels of low volatility

* "Diesel-Engine Exhaust Smoke as Influenced by Fuel Characteristics," by H. D. Young; "Smoke in High-Speed Diesel Engines," by J.J. Broeze and C. Stillebroer; "Influence of Injection Characteristics on Exhaust Smoke," by Martin Berlyn; "Rating Diesel Engines on Smoke Curve Basis," by Kenneth J. Fleck; "Technique of Exhaust Smoke Measurement," by Jordan P. Jung and Frank L. Coers; and "Operator's Viewpoint of Exhaust Smoke," by A.S. Leonard, were presented at SAE Annual Meeting, Detroit, Jan. 14, 1948.

took more time and mixing was not complete. With fuels of average or high volatility the effect was hardly noticeable. On the other hand, with fuels of very low ignition quality, the pressure rise came too late, hence, the higher consumption, and the larger quantities of soot released during the long delay period apparently could not be complete-consumed.

Many other precombustion engines show best consumption at full load with a fuel of optimum ignition quality. What cetane number this will be, whether it is also the optimum in other respects, depends on the peculiarities of the design being considered.

The main problem to avoid at low load is the intensive mixing of fuel with cold air and the deposition of fuel on cold walls. The least tendency to smoke is obtained if the fuel is prevented from dispersing in contrast to the requirements of full-load operation. The best, if not the only, way to reduce smoke is, of course, to keep the combustion space hot.

The latter cannot be done in a direct-injection engine, and so smoke is usually visible at fast idling. High volatility is favorable because such fuels have the least tendency to condense near the cool combustion-chamber walls. Ignition quality is of less importance unless ignition is excessively late, favoring dilution and deposition during the long delay period. Consumption is hardly affected by the loss of unburnt fuel, but is appreciably affected by the late pressure rise with low-cetane fuels.

Under idling conditions the influence of fuel properties in swirl-chamber engines is similar to that in direct-injection engines; however, owing to the hot swirl chamber, evaporation is quick and smoke is noticeably less dense. Again the major factor is volatility, although conditions become worse with delayed ignition, but this condition occurs only with fuels of rather poor ignition quality.

Conditions are more favorable for the formation of smoke at acceleration after a period of idling. For then the same quantity of fuel as under full load may enter a cold combustion space. In a direct-injection engine overpenetration cannot be avoided and the deposited fuel cannot come off the walls in time. The mixing process is upset, although it may be satisfactory under conditions of equilibrium. Thick smoke results, consisting of all possible smoke constituents, including unburnt fuel. It cannot be remedied by a hot combustion space because it is impossible to have the piston crown sufficiently hot when idling and not excessively hot at full load.

Engines with a heat-insulated precombustion chamber are not much troubled by the acceleration smoke problem. The high fuel-air ratio in the precombustion chamber causes its temperature to vary relatively little with load. During acceleration the mixing conditions are not much different from those at full load at the same momentary speed, and combustion remains unaffected.

High jacket temperature also appreciably reduces the smoking tendency of an engine, possibly overshadowing the fuel effect.

The injection characteristics of a direct-injection engine have considerable influence on smoking tendencies. First, there are certain mannerisms of injection that often (although not invariably) give rise to exhaust smoke and in addition are almost never found to be beneficial, so that they should be eliminated on general principles. These are: secondary injections, nozzle dribble, a long, declining tail on the fuel pressure-time diagram, inadequate initial rate of injection, and anything leading to protracted duration of fuel discharge.

In addition, there are other injection design problems that must be considered if smoking is to be held to a minimum:

1. Excessive penetration of spray; The desirable degree of penetration depends on the type of engine. In general, engines with a high degree of turbulence

in the air charge during compression prefer a soft spray of low penetration, whereas those having more quiescent air in the combustion chamber require considerable penetration.

2. Insufficient penetration: This condition causes smoke, particularly in open-chamber engines of the quiescent type. Smoke is not apparent at light loads, but becomes considerable at higher loads, even before maximum bmep is obtained (with reasonable air utilization.)

3. Uneven or improper dispersion: Some engines are very sensitive to this condition, but others are not - they will work with single-orifice, multi-orifice, plain-pintle, or throttling-pintle nozzles.

4. Unsuitable droplet size: Much research is needed on this subject. There is evidence that some highly turbulent engines prefer small droplets, some seem to thrive on uniformly small droplets, others like at least two different droplet sizes.

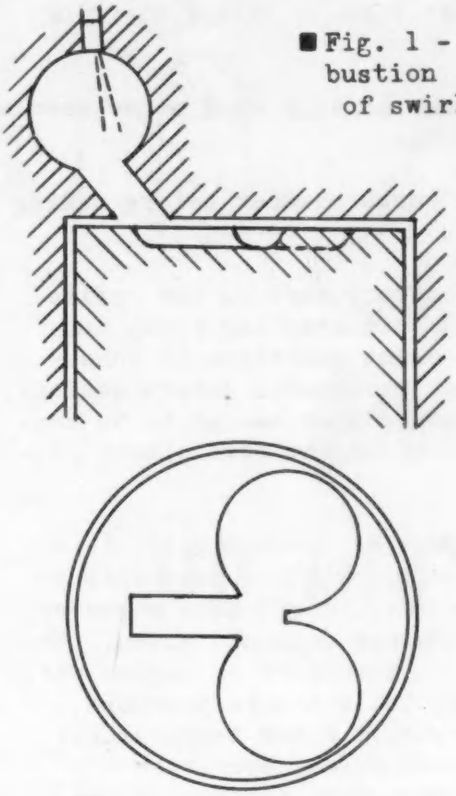
5. Improper timing of start of injection: This condition may lead to smoke at certain combinations of speed and load. Timing may also vary from cycle to cycle, causing intermittent, puffy smoke.

6. Improper rate of injection during discharge: Almost all diesels prefer a high initial rate of injection.

7. Excessive duration of injection: This condition is probably the most common cause of smoke. A short, crisp injection with a high initial rate of discharge, minimum fluctuations of rate, and quick cutoff without secondary dribble are needed.

The successful marriage of fuel and air in the cylinder is the joint responsibility of the engine designer and the injection engineer. Smoke-free operation under all reasonable conditions of service requires the closest collaboration of these two throughout the development of the engine.

A study of smoke-measuring techniques indicates that most consistent results are obtained when a sampling valve having a 1/8-in. wide slot is located in the exhaust pipe 43 in. from the exhaust manifold flange and connected to a CRC smokemeter by means of 72 in. of 3/4-in. OD, 1/16-in. wall copper tubing.



■ Fig. 1 - Precombustion chamber of swirl type

The following procedure for measuring smoke seems to give good results:

1. Scavenge the entire sampling system with compressed air to remove condensation after warming up the engine.

2. Set smoke density meter needle to zero. If adjustment is not sufficient to set zero, clean photocell lenses.

3. Open smokemeter throttling valve to give at least 8 in. of water charging pressure and leave valve open for 0.4 min.

4. Close throttling valve.

5. Read smoke density immediately.

6. Scavenge the smoke chamber before taking the next sample.

A simpler procedure, which is believed to be equally satisfactory, but on which further testing is recommended is:

Steps Nos. 1 and 2 are the same as in the first procedure.

3. Open smokemeter throttling valve to give at least 8 in. of water charging pressure.

4. Read smoke density when meter needle stops climbing.

5. Scavenge smoke chamber before taking next sample.

Further laboratory work on the refinement of the smoke sampling setup and smoke measurement technique is scheduled, so that smokemeter determinations may become dependable enough to be considered part of engine development programs.

The measurement of smoke density is most important, for it is desirable to include some measure of smoke whenever engine performance data are given. The smoke curve is a measure of engine performance, and tells a more profound story than exhaust stack temperatures. Proper maintenance and operation of trucks also have much to do with the amount of smoke the engine emits. For instance, if the driver tampers with the delivery adjustment on the fuel pump, or operates the vehicle improperly, rich, smoky exhaust may be the result.

One method of overcoming driver tampering may be to provide an internal stop limiting maximum delivery of the fuel pump, which almost nullifies any attempt to change delivery when the outside adjustment is changed. Driver training is also helping to teach drivers how to handle their equipment properly.

(Complete papers on which this article is based are available from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Steel Hardenability

Continued from page 48

Rockwell C 50 hardness. (For 0.40% carbon steels this gives 90% martensite.) Here the two steels with lowest hardenability - SAE 9440 and 1340 - will do for a 0.6-in. round when hardenability is the minimum specified.

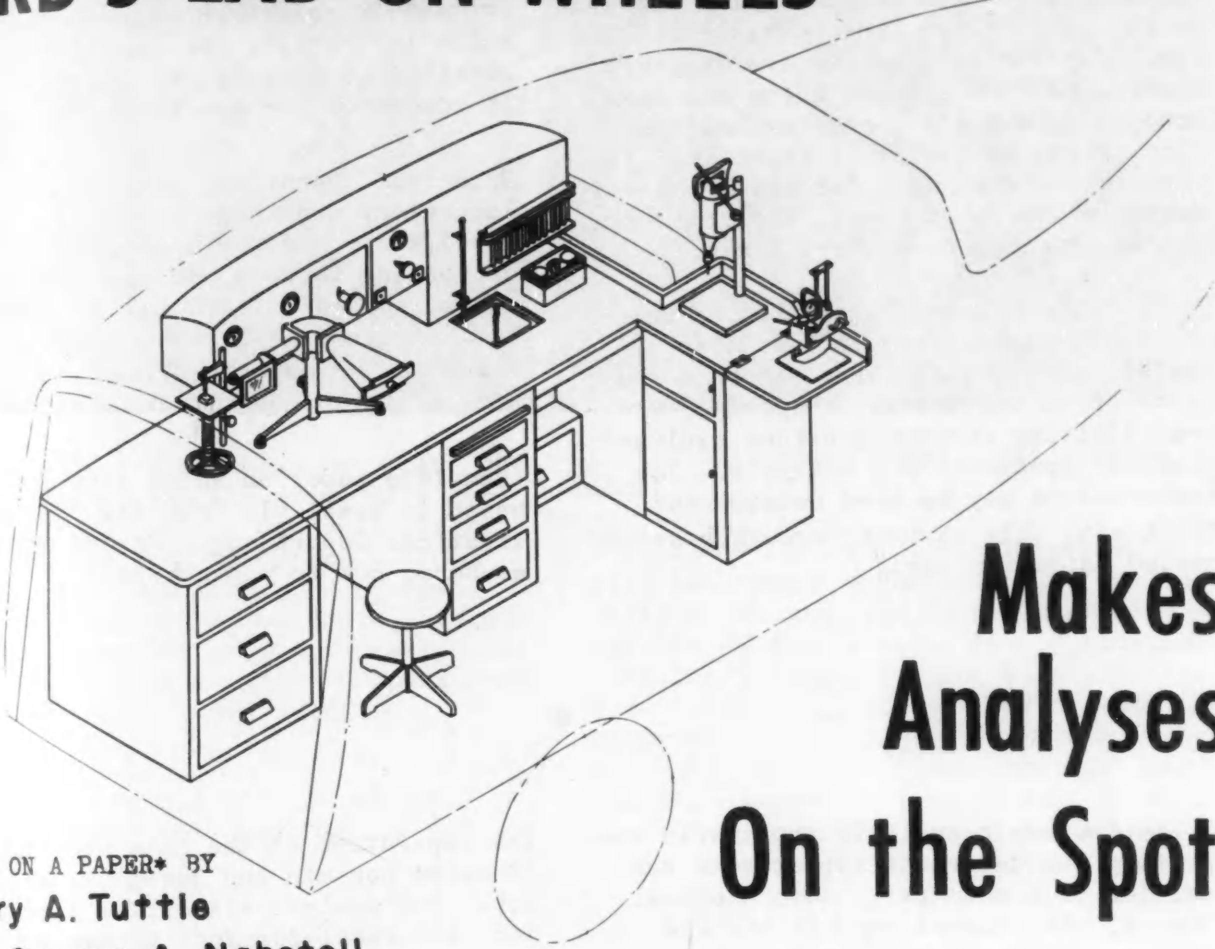
Engineers today are trying to decide whether steel-buying to chemical composition or to H-band specifications gives them the narrowest spread in hardenability. Fig. 12 compares the two methods. It shows minimum size that would harden to a depth of one-quarter radius to Rockwell C 50, the lowest hardenability with minimum chemical specifications shown on the right and the guaranteed minimum when buying to H-bands on the left. It's interesting to note that in several cases the lowest hardenability with minimum chemical composition matches that guaranteed in H-band buying.

Some metallurgists feel they gain very little on the low side of hardenability for some steels purchased by H-bands, especially since the low side of hardenability is the important one. It's possible to get a wide variation from the calculated average hardenability. Therefore, buying by H-bands would eliminate those heats lower in hardenability than the calculated average. This indicates an advantage on the low side of hardenability for H-band buying, although it may be small in some cases.

For steels SAE 1340, 4640, and 8640 the hardenability bands are much higher at the minimum than are calculated values. In almost all cases hardenability on the high side is lower using H-bands rather than chemical specifications. This is also an advantage for H-band buying. If steel producers could narrow the bands still further, it would be well to bring the low side up. Calculations show this may be possible.

(Complete paper on which this article is based is available from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

FORD'S LAB-ON-WHEELS



Makes Analyses On the Spot

BASED ON A PAPER* BY
Henry A. Tuttle
and George A. Nahstoll
FORD MOTOR CO.

FORD brings the mountain to Mohammed with its mobile laboratory that travels to various parts of the 1200-acre Rouge plant to make chemical and physical tests of metals in storage, scrap, and work in process.

Bringing the laboratory to the work place instead of sending a sample to a fixed laboratory saves time and increases flexibility of the plant's chemical engineering operation. By making on-the-spot tests of varied ferrous and non-ferrous materials, the motorized laboratory more closely integrates the chemical engineering department with production demands.

The truck itself is a standard 122-in. wheelbase Ford delivery panel type. A work bench contoured to body shape is built in on one side, as shown in the sketch above. Chemical test equipment consists of a semi-portable spectroscope, chemical spot-test kits, apparatus for carbon determination by color, and a portable high-speed grinder for spark testing. For physical tests the truck carries hardness testers of the Brinell, Rockwell, and Scleroscope types and a magnetic plate-thickness indicator.

* Paper "The Control of Materials by a Motorized Laboratory," was presented at SAE Annual Meeting, Detroit, Jan. 13, 1948.

A 3-kw generator, driven by a power take-off unit mounted on the truck-transmission, furnishes 220-v d-c electrical power. A 1700-rpm engine speed is enough to drive the generator at 1800 rpm. Electrical circuits are remotely controlled from a panel above the work bench. An auxiliary starter and ignition switch on the panel start and stop the engine. When used indoors or near a suitable source of power, the truck can be connected up to factory power.

Power tools can be plugged in for use inside the laboratory or used with a reeled extension cord for moderate distances from the truck. A special current-limiting circuit provides excitation for spectroscopic analysis. The spectroscope may be used outside the truck with this circuit through a second reeled extension cord.

All available space inside the truck is used without cramping the operator unnecessarily. More delicate equipment is protected against shock and vibration while in transit. As much equipment as possible is attached to the bench, but is removable for use in the field.

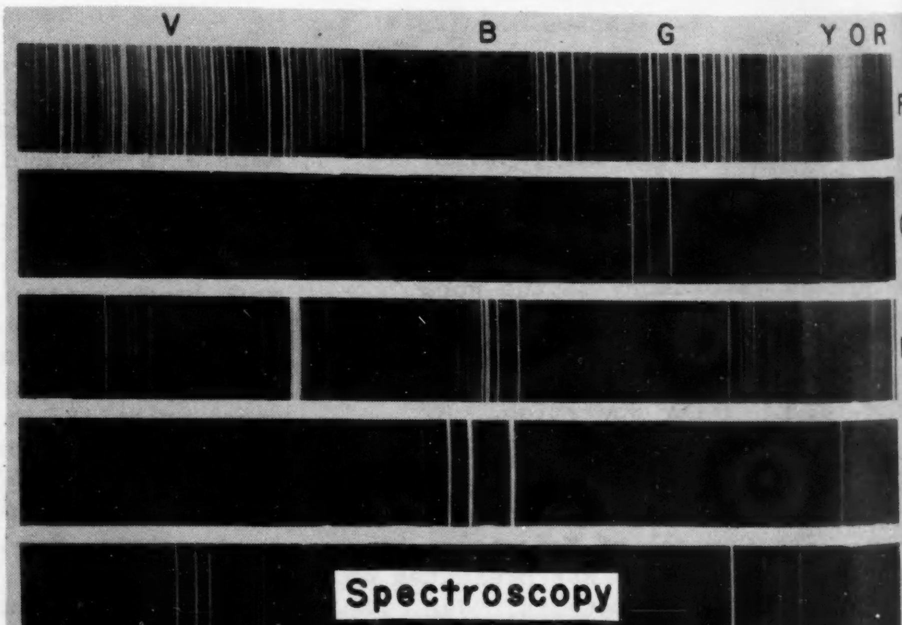
Principal operations performed by the laboratory are inspection or grading steel scrap, or raw materials, checking or sorting ferrous and non-ferrous stock, and identification or testing semi-finished or fabricated parts. Shown below is the motorized laboratory doing some of these jobs throughout the plant.

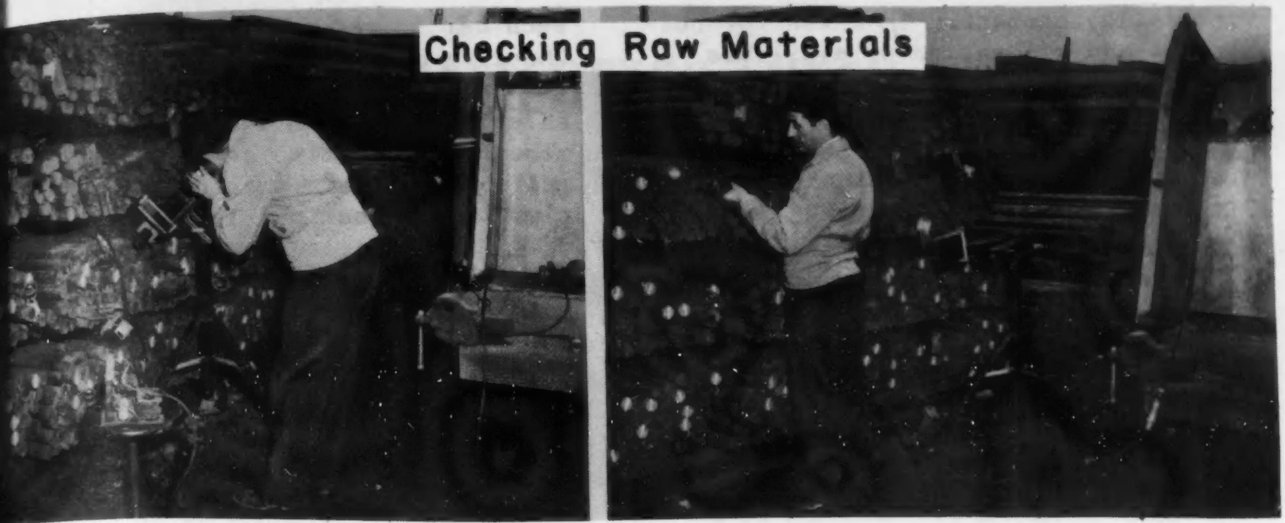
(Complete paper on which this article is based is available from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

■

By spectroscopic analysis the mobile laboratory can determine approximate composition of a material. Each chemical element, when viewed through the spectroscope, produces a characteristic emission of colored lines, as shown at right. Line intensity is roughly proportional to concentration of that element in the sample.

In one case the mobile laboratory's spectroscopic analysis untangled a mix-up involving 800 cluster gear forgings, some of SAE 8620 H and some of SAE 1030 steel. This non-destructive method made the separation at the rate of about 8 to 10 units per min and saved the entire lot. The analyst also found a number of SAE 5125 steel forgings in the mix-up whose presence wasn't even suspected.





Checking Raw Materials

The mobile laboratory also rectifies on-location mix-ups in the steel mill involving billets, bars, and coil stock. Here spark and spot tests are useful, although often spectroscopic equipment helps. The analyst shown above, at left, is using the spectroscopic equipment. At right he's shown conducting a spark test to separate mixed steel stock.

Making a separation or type analysis in the yard rather than sending the samples to the laboratory saves considerable time.

A four and one half month survey of mobile laboratory operation in scrap and steel stock analysis showed that 2685 samples were analyzed. It's estimated that the old way of doing it would have required 2876 hr against 768 hr with the motorized laboratory. (The former figure includes time to collect, prepare, and analyze samples.) This amounts to a labor saving of 475 hr per month.

■

Most valuable job done by the mobile laboratory is steel scrap checking and sorting. Scarcity of steel scrap since the war and use of mixed wartime scrap of questionable origin has hampered production of quality steel.

The motorized laboratory controls

Scrap Analysis

scrap alloy content. It makes for better use and allocation of available scrap, speeding a steady flow to mill and foundry. On-the-spot analysis beside a railroad scrap car, as shown at the bottom of p. 61, eliminates the need for sending samples to chemical and spectrographic laboratories. Spectroscopic classification into carbon, low, and high alloy steel is adequate.

Several scrap pieces are selected from each car to get representative analysis. Checking operation in the truck takes less than a minute per sample. The mobile laboratory on location makes for more representative sampling and more reliable analysis than does sending samples to the chemical laboratory.

Since initiation of this operation, demurrage charges on railroad cars due to delays in receiving laboratory reports have dropped to zero.

Incoming car, bus, and truck frames are checked periodically for dimensions at several inspection layout departments throughout the plant. The mobile laboratory is called in to check the hardness of component parts. This obviates the need for cutting up frames to send samples to the laboratory.

Finished Material Testing

Either the Ames Precision Hardness Tester or the King Portable Brinell is used, as shown in the operation at left. Nonferrous material - such as brass stock for radiators - can be checked quickly by spectroscope for type and impurities present. Hardness determinations are made with the Scleroscope. Metallic platings can be identified by the spectroscope and their thickness measured by the Magne-Gage.

The mobile laboratory can be dispatched wherever needed in case of trouble to determine the cause of failure and extent of damage. In this way it saves much production time and money.

But it is not always self-sufficient. Where an accurate chemical composition, specialized fatigue test, or salt spray test is required, all that the lab-on-wheels can do is to transport the sample to the main laboratory. When only semi-quantitative analyses or simple physical tests are needed, the motorized laboratory makes for materials control never before possible.

COORDINATING HELICOPTER PITCH AND POWER

Teamed by Linkage Design For Greater Flying Safety

FROM A PAPER* BY

A. F. Donovan

Head of the Aero-Mechanics Department

and Harold Hirsch

Head of the Helicopter Section

CORNELL AERONAUTICAL LABORATORY

SUDDEN large changes in rotor pitch - which a helicopter pilot must often make - result in sudden and dangerous changes in rotor rpm unless the throttle is readjusted at once.

Satisfactory simultaneous throttle adjustment can be secured by mechanically connecting the throttle control to the pitch control.

Reason why rotor rpm must be so carefully limited is that overspeeding stresses the blades and hub, while underspeeding brings excessive rotor coning and flapping angles. Most helicopters permit rotor speed variations of only 10 or 15% from the mean operating value.

The combination of this narrow rpm operating band and the particularly small flywheel effect of helicopter rotors requires that the torque required by the rotor and the torque delivered by the engine be kept nearly equal at all times.

The torque required depends largely on the rotor pitch. The torque delivered depends on the throttle position. Therefore, it is desirable to coordinate pitch and throttle settings so that torque delivered equals torque required at all times. Then changes in rotor pitch made in maneuvers are compensated for by changes in throttle setting, and rotor rpm remains constant.

Of course, changes in air density and flight conditions also result in variations in torque required. After such changes, the throttle must be reset to bring torque delivered in line with torque required. But these variations in torque required are smaller and occur more gradually than the torque variations due to changes in collective pitch setting, and throttle changes can be made manually or by means of automatic servos.

Good coordination of the pitch and throttle controls can be obtained by use of a dead-center bellcrank-link arrangement to linearize the parabolic pitch-torque relationship of the rotor and a cam for linearizing the torque-throttle relation of the engine. This arrangement makes possible the direct interconnection of the two control systems at one main pitch throttle control stick. The coordinated system can be designed so as to add negligible cost, weight, and complication beyond that required by independent pitch and throttle control systems.

*Paper "Helicopter Controls for Pitch-Power Coordination," was presented at SAE Annual Meeting, Detroit, Jan. 16, 1948.

To this combined system can be added automatic servos to compensate throttle setting for those changes in pitch setting which result from changes in air density and flight conditions.

WHY COORDINATION IS NEEDED

It is easy to understand why coordinated pitch and throttle regulation is needed when one considers how fast rotor rpm can change following a large change in rotor collective pitch.

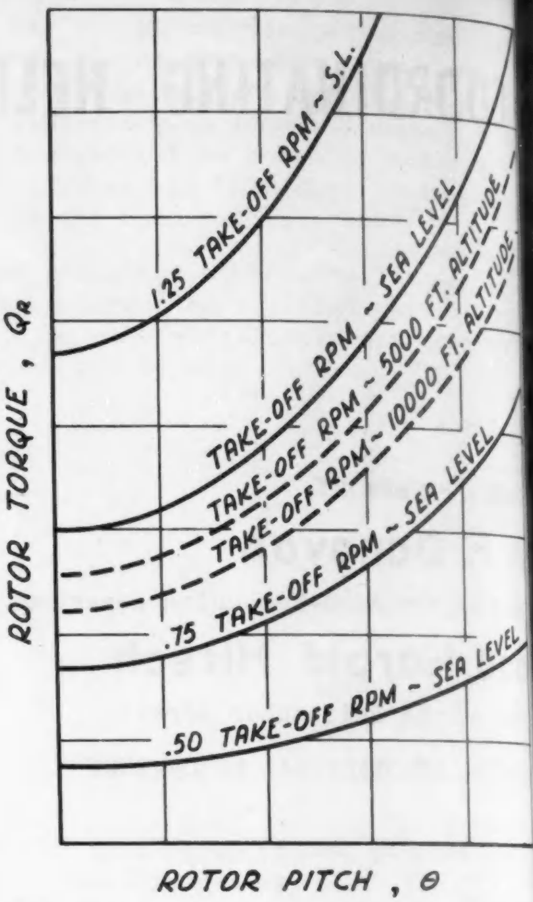
Consider, for example, the jump take-off of a 3000-lb helicopter having a 38-ft rotor, a rotor-system moment of inertia of 500 slug-ft^2, engine power rated at 240 hp at 2400 rpm, a rotor gearbox reduction ratio of 10 to 1, and a rotor speed of 240 rpm at rated engine power. Power required by such a helicopter when resting on the ground with the rotor blades at minimum pitch but turning at take-off rpm is about 40 hp. A sudden increase in rotor collective pitch such as used for a jump take-off will increase the power required by the rotor to the full rated 240 hp of the engine.

It can be shown that if the throttle setting is left unchanged, the rotor speed will drop from the initial 240 rpm to 107 rpm in 1 sec!

Since lift is proportional to the square of the rotation velocity, the decrease of roughly one half in rotor rpm causes lift to drop to roughly one quarter of its initial value. Obviously, a jump take-off executed without increasing throttle setting would result only in a sudden jump into the air - followed by a crash to the ground.

If the job of coordinating pitch and power is left to the pilot, he must devote the major part of his attention to watching the rotor tachometer and controlling the engine during maneuvers - when he most needs to watch helicopter position, attitude, and other external details.

A better way to accomplish coordination is to connect the collective pitch and throttle systems to the same operating



■ Fig. 1 - Rotor torque versus rotor pitch

lever and add a trimming adjustment for the throttle to compensate for changes in power needs. A number of coordinating systems have been devised, ranging all the way from simple linear connections to elaborate cam and link arrangements.

The trouble with some of the coordinating systems is that they do not correlate pitch and throttle setting properly at all points throughout the range of rotor pitch angles.

ACHIEVING COORDINATION

The way to plan a pitch-power coordinating system is to determine the rotor pitch-torque relationship and the engine throttle-torque relationship. This information can be combined to show the desired pitch-throttle relationship at constant rotor speed. Then the problem is to find a linkage which will coordinate pitch and throttle controls so that

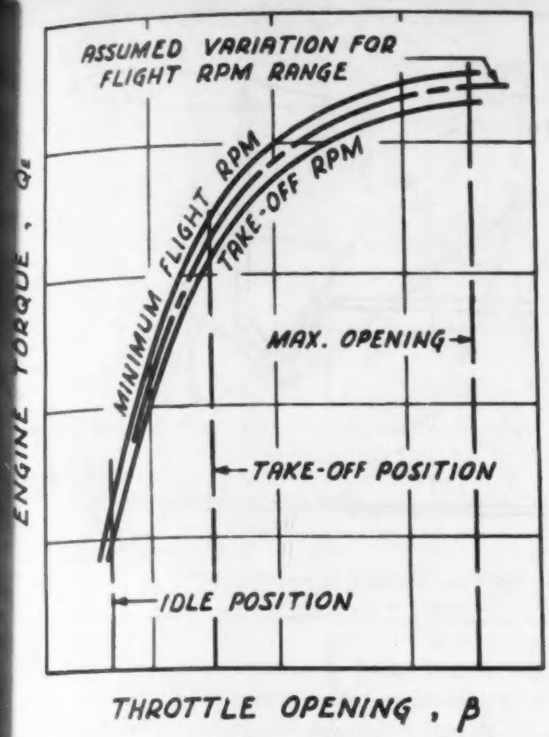


Fig. 2 - Engine torque versus throttle opening

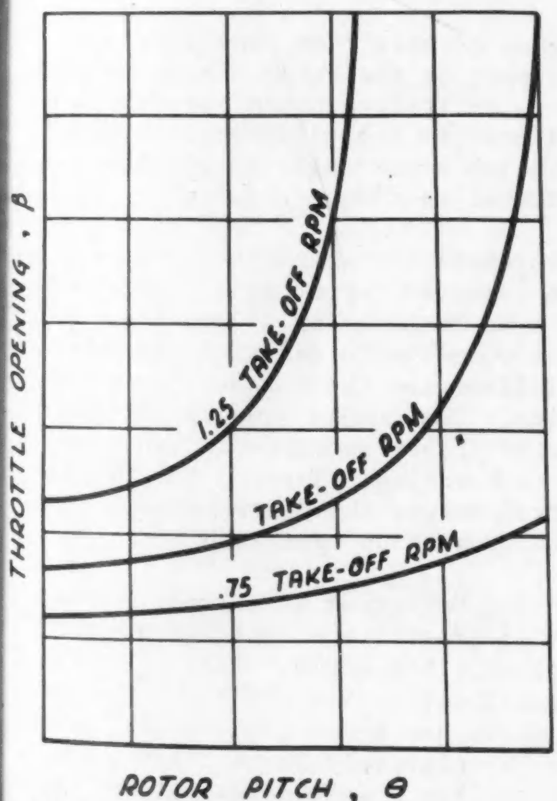


Fig. 3 - Throttle opening versus rotor pitch

the desired pitch-throttle relationship results.

Analysis of helicopter rotor thrust characteristics for constant altitude and rpm shows that for both vertical flight and forward flight the curves of rotor torque versus pitch are portions of parabolas. Fig. 1 shows typical curves for different rotational speeds and air densities.

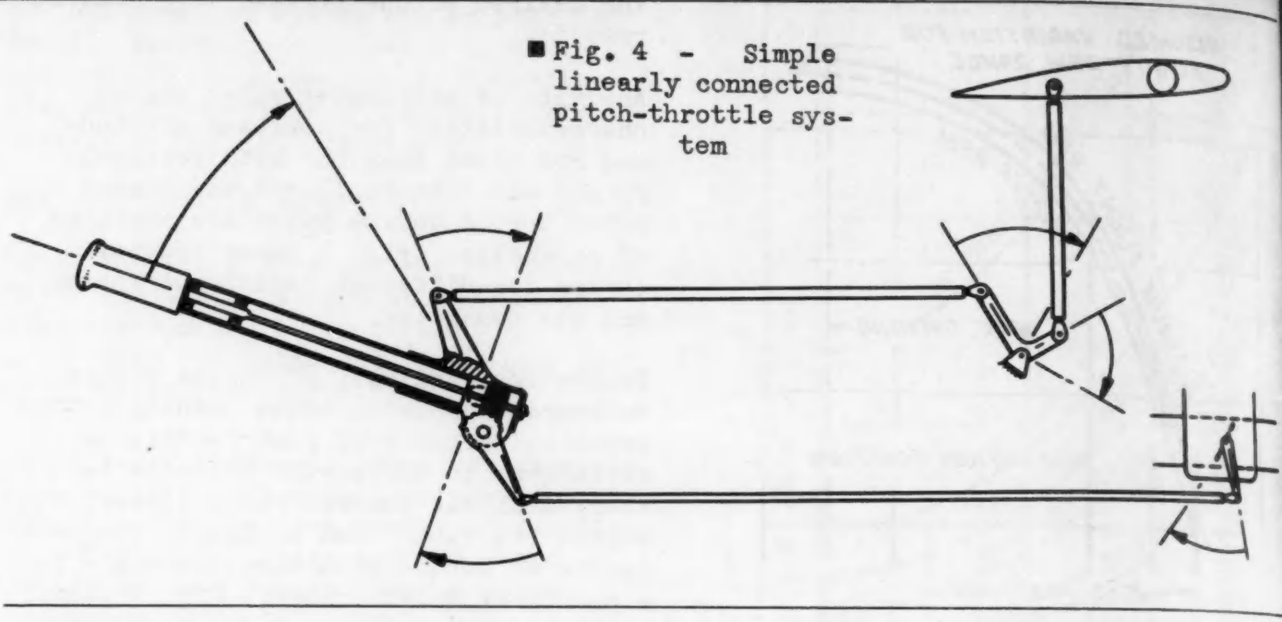
Theoretical analysis of engine torque delivered versus throttle opening is impractical because of such factors as variations in carburetor characteristics, manifold losses, valve losses, and engine friction. But certainly the variation of torque with throttle angle is a nonlinear relationship. Fig. 2 shows typical curves of torque delivered as a function of throttle angle. The exact shape of the curves varies from engine to engine and can be determined experimentally.

By reading off torque values corresponding to selected rotor pitch values on a curve of Fig. 1, and reading the values of throttle opening on Fig. 2 corresponding to those torque values, it is possible to plot a curve of throttle opening versus rotor pitch, like those in Fig. 3. This curve shows how coordinating control should vary throttle setting with pitch setting to keep rpm constant.

Since the torque required by the rotor and the torque delivered by the engine are both functions of air density, density disappears as a factor when the two torques are equated. Therefore, a pitch-throttle coordinating system designed to hold rpm constant at sea level will also hold rpm constant at altitude (providing an unsupercharged engine is used).

COORDINATING SYSTEMS

First attempts at a coordinating system resulted in a mechanically simple, linearly connected system like that diagrammed in Fig. 4. The correlation it gives is shown in Fig. 5. Obviously, correlation is good only at two pitches, one high and one low. At all other pitches, the manual throttle trimmer must



■ Fig. 4 - Simple linearly connected pitch-throttle system

be readjusted to hold desired rotor rpm.

Rotation of the throttle trim handle also produces linearly proportional motion of the throttle, giving a typical power versus throttle motion relation. If rotor speed is changed to a new value by rotation of the manual throttle grip, operation of the collective pitch lever will still produce linear throttle operation. If plotted on Fig. 5, the rotor pitch-throttle opening curve would be a straight line parallel to the original straight line. Since throttle motion is proportional to handle rotation at all rpm's and collective pitch settings, the variation in rotor driving torque per unit rotation of the throttle grip is proportional to the slope of the throttle-torque curve, Fig. 2.

If rotor rpm is held constant by use of the throttle trim control, operation of the collective pitch control results in a change in rotor lift directly proportional to the pitch handle movement.

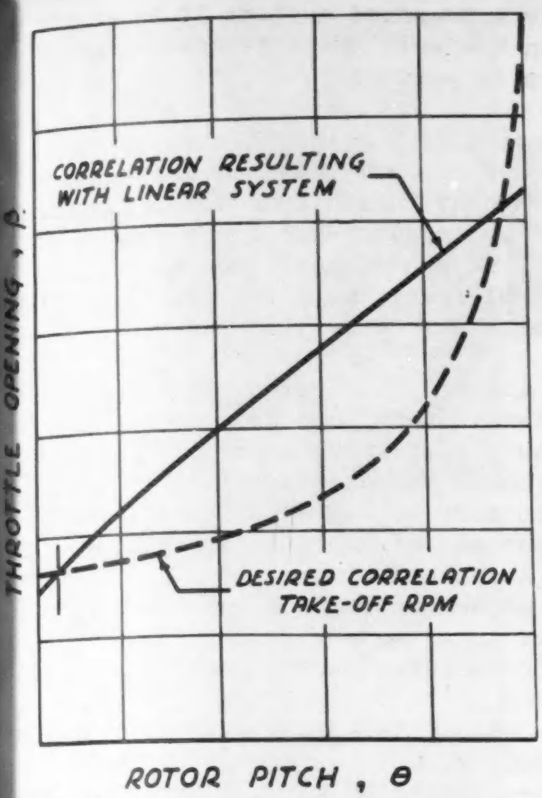
In the evolution of pitch-power coordination systems, improved designs have appeared which give much better correlation than the early linearly connected systems, without greatly increasing the mechanical complexity. Fig. 6 shows a proposed design, called the dead center pitch system, which uses a bellcrank arrangement in the pitch system to linearize the parabolic pitch-torque relation of the rotor and a cam for linearizing the torque-throttle relation of the engine. Fig. 7 shows the correlation obtained.

It can be shown that, because of the special bellcrank near the rotor, pitch stick displacement is proportional to the pitch angle squared.

Because of this fact, and also because that part of the rotor torque required due to collective pitch increase is proportional to the pitch angle squared, the pitch stick angle is directly proportional to torque required.

Measurement of the pitch-throttle relation required for constant rpm's is easily made in hovering. From these data, it is possible to design a cam which will linearize the torque-throttle relation. The cam is usually designed to provide proper coordination at take-off rpm in hovering. Turning the throttle control shifts the design curve horizontally without changing its shape.

With the bellcrank arrangement, rotor lift is proportional to the square root of pitch stick angle. This means that at small collective pitch stick angles, the pilot has a small mechanical advantage; a relatively large pitch change results from a unit stick motion. As the pitch stick angle increases, so does the pilot's mechanical advantage; small-



■ Fig. 5 - Correlation provided by simple linearly connected coordination system

er rotor pitch changes result from each unit pitch stick motion.

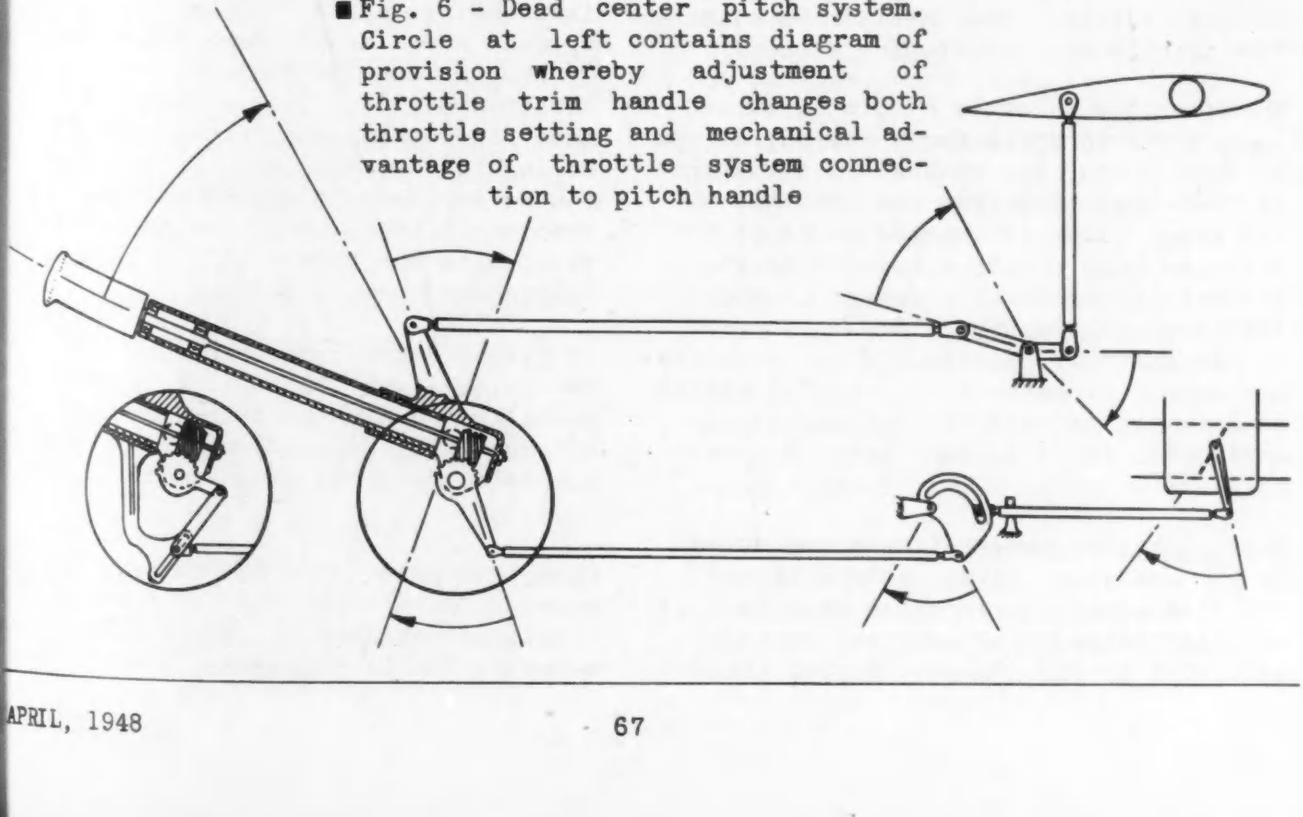
This arrangement has two important advantages:

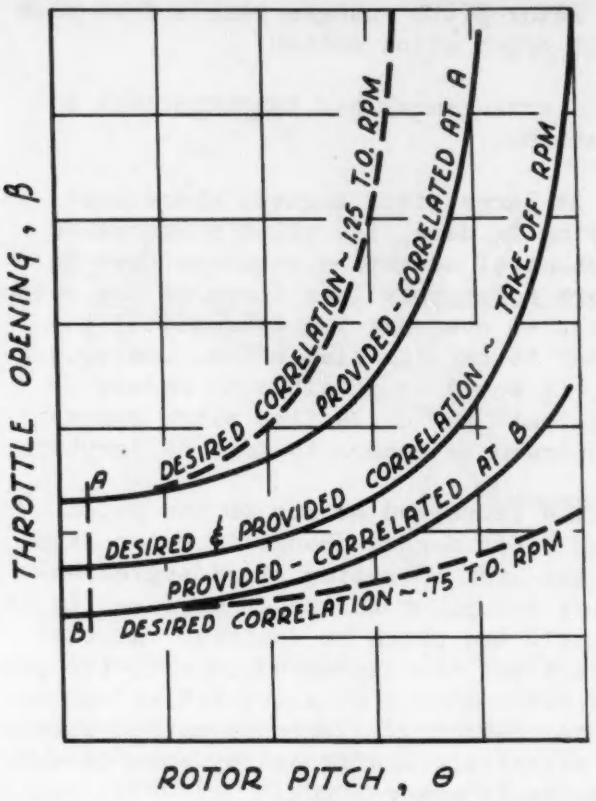
1. At large pitch angles, where most flying is done, the pilot's increased mechanical advantage requires that he exert relatively less force on the pitch stick to overcome the tendency of the rotor to go into flat pitch. Consequently, it would be possible to reduce or eliminate the collective pitch bungee requirements common to most helicopters.

2. The increased motion of the pitch handle per degree change in rotor pitch at the high operating pitch angles enables the pilot to adjust more easily to exactly the pitch he desires. At the same time, the increased sensitivity of the pitch handle at autorotative angles is not detrimental because autorotation is relatively unaffected by considerable changes in rotor pitch.

One further improvement which can be made in the system is illustrated in the circle at the left in Fig. 6. With this arrangement, adjustment of the throttle trim handle changes both the throttle setting and the mechanical advantage of

■ Fig. 6 - Dead center pitch system. Circle at left contains diagram of provision whereby adjustment of throttle trim handle changes both throttle setting and mechanical advantage of throttle system connection to pitch handle





■ Fig. 7 Correlation provided by dead center pitch system

the throttle system connection to the pitch handle. Rotation of the throttle grip pushes the connecting link to the throttle along the slot which runs down in the pitch-handle throttle bellcrank. Twisting the throttle handle at low pitches advances the throttle by sliding the link rod in the sloping slot.

Whenever the throttle handle is turned to set the throttle for a new engine rpm at zero pitch, the mechanical advantage of the connection from the throttle to the pitch lever is changed to cause the rotor to hold constant speed when the pitch is changed. The change in mechanical advantage causes the displacement of the throttle cam input link to follow the family of parabolas in Fig. 1 rather than to approximate all but one by a horizontal shift of the take-off parabola.

Barring higher order effects neglected in the analysis, this system with mechanical advantage throttle trim fulfils ideally the coordination requirements specified by Fig. 3. Of course, this is only a proposed system; it remains to be seen how well the arrangement will perform in practice.

Trimming of the throttle can be accomplished manually, as has been assumed throughout this discussion, or it can be done by an automatic speed-sensitive servo or a governor. Automatic speed control frees the pilot from concentration on the rotor tachometer.

The linearly connected pitch-throttle system is very poor from the standpoint of automatic servo control. The coordination is so unsatisfactory that the servo must be almost as fast as though no mechanical coordination were used and the servo had to provide all the throttle adjustment. Such service would require an extremely fast servo - which would be large, heavy, and expensive.

The dead center pitch system, with or without mechanical advantage trim, is better suited to servo control. Because of the excellent coordination produced mechanically, relatively slow and inexpensive servos can be used to produce close rpm control. The servo should be installed between the pitch stick and the throttle linearizing cam so that the engine torque at constant rpm will be linearly proportional to the servo extension. This can be done quite easily by inserting an electrically operated irreversible screw jack in the system at a point whose motion is normally linearly proportional to that of the pilot's throttle handle. Provision should be made for locking the throttle handle during servo operation. Or the servo can be arranged to engage and turn the throttle handle during automatic operation, with the pilot following the servo-produced handle motion.

If flight tests prove that the dead center pitch system is as good as it appears from analysis, that system, with automatic throttle trimming, will offer complete, reliable pitch-power coordination.

(Complete paper on which this article is based is available from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

TECHNICAL COMMITTEE PROGRESS

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SAE To Classify Oils

For Auto Transmissions

JOB of classifying by viscosity hydraulic fluids for torque converters and fluid couplings in automatic transmissions is being tackled by the SAE Fuels & Lubricants Technical Committee.

Automatic transmission specialists feel viscosity influences torque converter efficiency because of the continuous power transmission, but not so much as do torque ratio and fluid temperature. Viscosity, they say, doesn't appear to be critical with fluid couplings.

Preliminary discussions in a subcommittee of the main group, headed by G. A. Round, Socony-Vacuum Oil Co., Inc., pointed up areas of general agreement within its membership. For one, the subcommittee indicates that suitable oils within the SAE 10, 20, and possibly 30 range will satisfy requirements of torque converters in which the power-transmitting fluid also lubricates transmission gears. These oils should also serve fluid coupling needs.

For converters with fluid acting only as the hydraulic medium, and possibly to lubricate seals, the conferees feel two or at most three fluids will handle viscosity requirements.

One of these might have the viscosity range of the contemplated 5W classification for low-viscosity oil for low-temperature performance - about 80 to 120 SUS, depending on viscosity index. They visualize the next lighter fluid as one with a viscosity of 45 to 55 SUS at 100 F. A still lower viscosity oil may be needed for such automatic transmission units in cold-weather service.

The subcommittee feels manufacturers of these units using lighter fluids must reach closer agreement on what's needed before any definite classifications can be proposed. But it's optimistic over possibility of early accord within the industry.

When ready, the viscosity classification set up by SAE will be referred to ASTM Technical Committee D-2, which is writing specifications for torque converter fluids.

Serving with Chairman Round on the subcommittee are H. C. Gooch, Tide Water Associated Oil Co.; J. D. Klinger, Chrysler Corp.; A. H. Deimel, Dana Corp.; C. J. Livingstone, Gulf Research & Development Co.; J. M. Miller, Standard Oil Co., (Ind.); R. N. Neldon, American Blower Corp.; A. C. Pilger, Jr., Tide Water Associated Oil Co.; W. F. Shurtz, Twin Disc Clutch Co.; and H. R. Wolf, General Motors Corp.

Airline Tests Gage Spark-Plug Erosion

HEAVY-ELECTRODE spark plugs for aircraft engines erode faster than fine-wire plugs during early operation, but eventually approach the wire-electrode erosion rate. These results, from tests conducted by Northwest Airlines, were described at the last meeting of the SAE Ignition Research Committee.

Six fine-wire electrode plugs and six heavy-electrode plugs were installed on six cylinders of a Pratt & Whitney R2000 engine operating in a DC-4 airplane. The two plug types were paired at each cylinder to get comparative erosion rate data. Both plugs were set with an initial gap of 0.008 in.

Observations were made during the full test period - the first check after 87.54 hr of operation, the second after 322.5 hr, and the third after 632.46 hr. The heavy-electrode plugs were run for a long period; the fine-wire ones were removed at the end of the normal 370-hr plug-removal period. They were replaced with newly-conditioned plugs to insure ignition, even, if the heavy-electrode plug's gap became large enough to cut out at take-off.

After the first test period, heavy-electrode plug gaps average 0.013 in. and those of the fine-wire plugs, 0.011 in. This amounts to erosion rates of 0.00569 and 0.00341 in. per 100 hr, respectively. Erosion rates at the end of the second period were 0.00319 for the heavy electrode and 0.00306 for the fine-wire ones. The heavy-electrode erosion rate at the end of the third period averaged 0.00284.

These test data point to higher erosion rates for the heavy-electrode plug during initial operation; but in time both plug types erode at about the same rate.

According to C. E. Swanson, Northwest Airlines, Inc., who reported these tests to the Committee, this work will be continued using more plugs.

Chairman A. L. Beall, Wright Aeronautical Corp., advises that the Committee is studying the possibility of standardized plug-rating procedures. Among other projects being tackled by the Committee, he said, are improved plug-testing methods and standardization of turbine-ignition plugs.

The Committee also is studying developments on high and low tension cables. It finds that silicones have potentialities in the aircraft and automotive industries because of their exceptionally high and low-temperature stability and oil and moisture resistance.

Philadelphia Section Joins Annual Helicopter Forum

The fourth Annual Helicopter Forum is being held at the Franklin Institute in Philadelphia, Pa. on April 22, 23, and 24. The sessions during the first two days will comprise of papers on helicopter configuration designs, mechanical design, operation, testing, maintenance and economics.

The last day will be devoted to Helicopter exhibits and demonstrations. Joint sessions are being held on April 22 and 23 with SAE and the Institute of Aeronautical Sciences.

Probe Polaroid Glass For Rating Bus Smoke

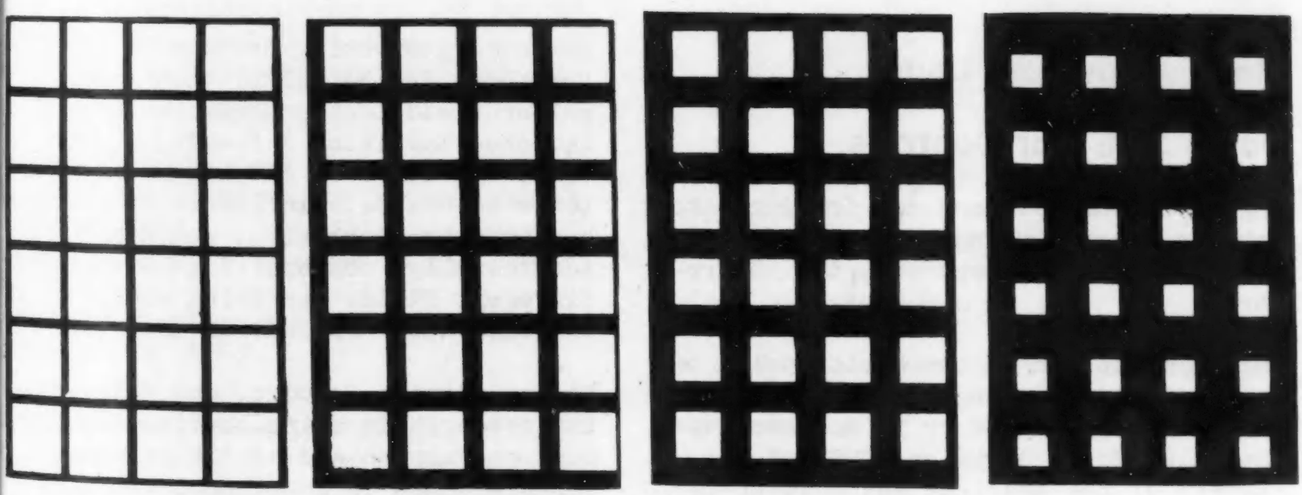
POSSIBILITIES of developing a polaroid optical device for measuring density of smoke emitted by trucks and buses are being investigated.

In its search for a satisfactory smoke-measuring method for law-enforcing officers, the SAE Automotive Exhaust Smoke Subcommittee has discussed the feasibility of a simple optical instrument. One suggestion is a device consisting of two polaroid lenses rotating over each other, fitted with a scale adaptable to smoke measurement. When the polaroid lenses are turned to give a degree of opacity matching the smoke observed, the reading on the polaroid lens scale could be taken as the measure of smoke density.

G. F. Roddewig, GMC Truck & Coach Division, will discuss this proposal with polaroid engineers.

A current smoke-measuring method uses the Ringelmann chart, a sample of which is shown in Fig. 1. When seen from 50 ft, the distance prescribed for rating smoke density with this chart, the four kinds of black-and-white areas blend into different shades of grey. Officials in Los Angeles now are using a variation of the Ringelmann chart; but others see drawbacks to its use.

For one, variations in printing could make substantial differences in values of the same number from one chart to another. Wear on the chart being carried in a pocket or brief case also would change intensity of the printed impression.



1. 20% BLACK 2. 40% BLACK 3. 60% BLACK 4. 80% BLACK

Fig. 1 - Ringelmann Chart used to measure smoke density. At a distance of 50 ft the squares blend into tones of grey, against which smoke is matched

Specify Rubber Cups

For Truck Brakes

NEW automotive rubber specifications are under way for both heavy-duty hydraulic brake cylinder cups and vehicle floor mats.

Current standard for rubber brake cups, issued in 1945 by the Joint SAE-ASTM Technical Committee A on Automotive Rubbers, is considered inadequate by the Committee for heavy-duty truck service. The cup defined in the present standard doesn't hold up under the heat developed by truck brakes.

For this reason the group intends to supplement the current specification, or to prepare a new one, spelling out performance tests and standards for heavy-duty type brake cups. The proposed specification, already in draft form, is intended for high temperature-resistant natural and synthetic rubber.

The Committee is also well along the road toward adopting a first-time standard for automotive rubber mats. It's to be a specification outlining physical test procedures and physical property requirements.

Many Amputees Using

Basic SAE Car Controls

NEARLY 35,000 veterans are driving automobiles specially equipped with aids to help disabled and amputees, the AMA reports.

During World War II the basic design of such mechanical aids was developed by an SAE committee headed by L. A. Chaminade, Chevrolet Motor Division, GMC, at the request of the American Association of Motor Vehicle Administrators.

A preview, with amputees driving various makes, was held under the sponsorship of Surgeon-General Kirk at Walter Reed Hospital.

Modifications of the SAE basic controls are made by each manufacturer. Some motor vehicle administrators report that disabled veterans, who receive special instructions with their new cars, have a much better safety record than the average motorist.

Current CRC Program

Envelopes Broad Area

COORDINATING Research Council's program for 1948 embodies nearly 80 active projects on common automotive-petroleum problems. Emphasis continues on engine-fuel-oil relationships in both the motor vehicle and aviation projects.

One big job in the motor fuel area, under the Coordinating Fuel and Equipment Research Committee, is the correlation of ASTM and Research octane ratings with road results. The group is working on a suitable road-testing technique. In the other direction on detonation, the Committee is aiming for a single-cylinder laboratory engine test reasonably duplicating road antiknock performance of commercial fuels.

Another interesting project in the motor fuel area is the sulfur test work. A sulfur field test program has been launched involving 7 fleets, 62 vehicles, and about 1, 500,000 miles of tests to evaluate influence of sulfur on engine wear, corrosion, and deposits. Additionally, commercial gasolines from different fields are being analyzed in the laboratory for sulfur content.

Engine varnish, sludge, and valve-burning research is being continued. A report on varnish and sludge research technique now is near completion.

Recent undertaking in aviation fuels research is a broad program to get the best fuel and gas turbine engine combi-

ation for civil aircraft flying under various temperature, load, and altitude conditions. To develop prototype fuels for this work, a group under the CFR Aviation Fuels Division is investigating behavior of gas turbine fuels in existing turbine fuel systems under simulated flight conditions.

Another group is concentrating on development of a better method of measuring antiknock quality of aviation fuel for piston engines. A report on the subject is now in the making. Another group is seeking to remedy the shortcomings of the present octane rating scale for aviation fuels. It has proposed a "detonation index" which can be extended indefinitely.

Among diesel fuel projects now under way is one to study effects of fuel-oil deterioration products on engine performance. This new effort will be concerned chiefly with storage stability of diesel fuels, laboratory research techniques for predicting it, and techniques for determining "so-called" gum. From information and methods evolved, the group handling this project hopes to correlate properties of various diesel fuels.

Lubricant researchers under CRC are cooperating with Army Ordnance cold-weather testing. This program is designed to help bring about petroleum products and equipment mutually suitable for operation at temperatures down to -65 F. CRC served for two- and three-week periods during this past winter at Fort Churchill and in Alaska.

Another job being done for Army Ordnance by CRC is rust-prevention for engines in long-term storage - possibly ten years. One of the objectives here is the perfection of a preservative insuring protection at storage temperature ranging from 135 F to -85 F.

Among several gear oil projects is one to determine moisture-corrosion characteristics of universal gear lubricants. The group assigned to this job will determine what constitutes injurious corrosion. It feels this will involve evaluation of various research techniques for determining moisture corrosion characteristics both of lubricants and equipment.

These are just a few of the nearly 80 projects currently under CRC aegis; but they demonstrate the nature of the overall program.

New Officers

Elected By CRC

J. M. Crawford, R. D. Kelly, Arthur Nutt, and J. C. Zeder, nominated by the SAE, and W. M. Holaday, K. G. Mackenzie, G. G. Oberfell, and H. G. Vesper, nominated by the API, were elected for two-year terms as directors of the CRC at the annual meeting March 4 in New York.

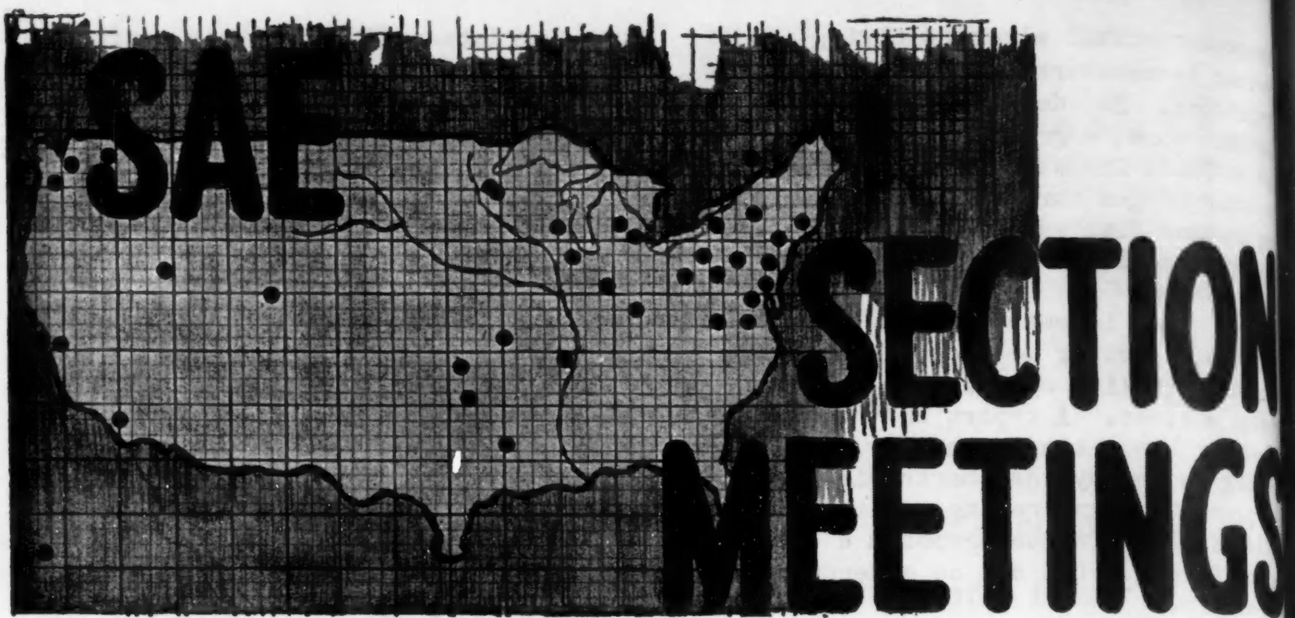
The board elected D. P. Barnard as president, C. E. Frudden vice-president,

D.P. Barnard, newly elected president of CRC

Genevieve Walmsley secretary, and Lacey Walker treasurer. W. T. Gunn and T. B. Rendel were elected assistant treasurers. M. K. McLeod was reappointed as manager for the current year.

The 1948 Executive Committee of the board consists of B. B. Bachman, D. P. Barnard, J. M. Crawford, C. E. Davis, C. E. Frudden, W. M. Holaday, R. D. Kelly, and K. G. Mackenzie.

A Finance Committee consisting of Messrs. Mackenzie, Bachman, and Frudden was appointed, with the former chairman.



The excellent reports of SAE Section Meetings written by our field editors have been rewritten for brevity because of the strike of typesetters. This less-than-satisfactory method of presenting the news of Section Meetings is the best we can do under the circumstances.

Reports of 14 SAE Sections, Groups, and Student Branches received up to press time covered a range of engineering topics almost as broad as the entire scope of the Society itself.

Current gasoline shortages gave timeliness to meetings of the FRESNO Division and PHILADELPHIA Section, when D.N. Harris projected the supply picture in terms of aircraft fuels on Feb. 2 at the former city, and William M. Holaday, Socony-Vacuum Oil Co., Inc., brought his 1948 Annual Meeting paper on The Efficient Utilization of Fuels up to date before the Philadelphia audience on Feb. 11.

Both agreed that price controls on crude and state restrictions on new well drilling were contributory causes, but use of petroleum products has been outstripping forecasts.

Refining research has made great strides during the past decade, and both speakers predicted that further efficiencies are ahead. Both pointed to the excessive cost per gallon of gasoline if shale and low grade coal are resorted to to augment supplies. Cost of initial equipment, they agreed, would be high.

Chassis dynamometers as a tool to reduce maintenance costs were described by Henry T. Mueller, Ethyl Corp., at the TWIN CITY Section on Feb. 5, and M.R. Bertolette, Clayton Mfg. Co., before the SALT LAKE CITY Group on Feb. 9

The former explained how money is being saved by the St. Louis Public Service bus lines by using this equipment. Maintenance processes have been greatly modified, he said, on the basis of experience with this equipment.

Savings from $125 to $150 per bus per year resulted from dynamometer testing which led to improved carburetor maintenance. Driver complaints dropped 12 to 15%, and road calls were reduced about 20% per vehicle.

Both speakers predicted wide use of this equipment as fleet operators understand its value.

Air transport and other aeronautical

topics were brought to the attention of BALTIMORE Section on Feb. 12, by Capital Airlines' president, James H. Carmichael, who reviewed the rapid growth of the industry. Dr. Hugh L. Dryden, new director of aeronautical research for NACA, explained the agency's scope in basic research for industry and the armed forces before METROPOLITAN Section on Feb. 18. WESTERN MICHIGAN Section was told of the role of the CAA at a Student Meeting, and the CALIFORNIA INSTITUTE OF TECHNOLOGY Student Branch sponsored an inspection tour of airline equipment maintenance.

Although railroads have been successful in consolidating their ticket sales, and the idea seems theoretically sound to Carmichael, he held that the time has not yet come for this move on the part of airlines. He is convinced that instrument flying will be commonplace within this decade.

Dryden agreed with a number of discussers that continued research on propellers is essential, and assured his audience that this is being given high priority by the NACA.

Caltech students went to Burbank to inspect the aircraft engine and propeller maintenance work of the Pacific Airmotive Corp., which has put equipment overhaul on a factory production line basis.

The case for synthetic rubber, held by J.E. Hale, Firestone Tire & Rubber Co., to be our nation's "best paid-up insurance policy," brought out a large NEW ENGLAND Section audience in snowbound Boston on March 2.

One-third of the rubber used in the United States is synthetic, and he sees a bright future for this material as chemists learn more of the science. Manufacturing difficulties have been overcome to a large extent.

Government action to keep a substantial tonnage of synthetic rubber in production is hoped for by the industry, because this will tend to improve product quality and lower cost.

Suspension was the topic on Feb. 9 at the UNIVERSITY OF BRITISH COLUMBIA Student meeting when N.E. Hendrickson, U.S. Spring & Bumper Co consulting engineer, described the large static displacement required to effect a good ride. Also consultant for the Vanadium Corp. of America, the speaker described the spring qualities of various alloys, which are lasting up to five times as long as carbon steel units.

A previous meeting of the Student Branch on Jan. 12 heard Robin L.I. Fjarlie, a student, tell of an engineering project in oceanography, in which a scale model was constructed of an area where a river would carry wastes from a proposed wood pulp mill, in order to study the effect of local tides.

BUFFALO Section heard a detailed description of the Playboy car, a 2-passenger, 48-hp, 2035-lb vehicle with a 90-in. wheelbase. The company's chief engineer, Charles D.Thomas, explained that it was designed to meet the demand for a small vehicle as a second car in the family, or for young people who had no need for large automobiles. Features are its welded integral body, and an all-steel convertible top.

A symposium on Engineering Education was presented Jan. 26 before CENTRAL ILLINOIS Section by Conrad E. Schwager, student at Bradley University, Prof. N.A. Parker, University of Illinois, and Glen V. Edmonson, who had been with the Hydraulic Division of American Blower Corp.

The student's viewpoint as presented was that industry prefers beginners who fully understand the requirements of the prospective employer, men who have a good fundamental grasp of engineering, creative ability, and some understanding of psychology and economics.

Parker reported that engineering schools have developed programs of organized cooperation with students for teaching products, company policies and organizations, methods of manufacture, and other pertinent subjects.

Speaking from industry's point of view, Edmonson, now on the faculty of the University of Michigan, held that the three characteristics sought in a young engineer by industry include presentable appearance and a good personality, good mind and serious outlook, and completion of four years of undergraduate engineering training. Ability to express himself was emphasized.

SOUTHERN CALIFORNIA Section members on Feb. 26 heard J.C. Miller, Cummins Engine Co.'s research manager, describe why crankshafts fail, and how to overcome these troubles through understanding torsional vibration.

He pointed out that these vibrations occur at the front end of the shaft which alternately rotates faster and slower than the steadier rotation of the rear end at the flywheel.

METROPOLITAN Section heard the inside story of two-way radio at first hand on Feb. 25, when Gordon R. Creecraft and his associates from American Telephone & Telegraph Co. presented that subject at a Panel Meeting.

Slow expansion may be expected, it was said, because of large waiting lists of applicants, shortage of two-way radio equipment, and scarcity of radio frequency channels.

AT&T charges $22 a month for general urban service. This includes $15 for equipment rental and a $7 service charge for a total of 20 three-minute connections from the vehicle within the local service area. Long distance charges will be billed. Installation is difficult on current cars, and costs about $25 per vehicle, besides the heavy-duty electrical equipment needed.

DETROIT Section members heard Andrew Kalitinsky, chief engineer of Fairchild's NEPA Division at Oak Ridge, describe in layman's language the physicist's concept of atomic power on Feb. 2.

Engineers must, he asserted, take an active part in putting this energy to work in powerplants, relieving scientists for more research. He raised the curtain on some of the phases or harnessing this energy, and is confident that further research will disclose other than now known materials to disintegrate.

Energy and heat developed by an atomic reactor might be applied to the steam turbine, turbojet, ramjet, and the nuclear rockets, he said.

SAE NATIONAL MEETINGS

MEETING	DATE	HOTEL
SUMMER	June 6-11	French Lick Springs, French Lick, Ind.
WEST COAST	Aug. 18-20	St. Francis, San Francisco
TRACTOR and DIESEL ENGINE	Sept. 7-9	Schroeder, Milwaukee
FUELS AND LUBRICANTS	Nov. 4-5	Mayo, Tulsa

ABOUT SAE MEMBERS

WILLIAM S. JAMES, director of research of the Ford Motor Co. since 1945, was elected vice-president in charge of engineering of the Fram Corp., manufacturers of oil, gasoline, and air filters. James is a past-president of the SAE and engineer in charge of the development of the Amphibious Weasel for the Army during the war. He resigned from Ford last month and will take up his new duties immediately. He is currently chairman of the SAE Publication Committee.

President of Studebaker Corp., PAUL G. HOFFMAN, has resigned as chairman and treasurer of the Automotive Safety Foundation. He played a major role in the establishment of this organization and served as its president from 1937 to 1942. Hoffman will continue to serve on the ASF Operating Committee.

The Heil Co., in Milwaukee, has announced that ARNOLD F. MEYER, chief engineer of the company for many years, has been made vice-president in charge of engineering.

ROBERT G. GREEN, formerly a project engineer at Packard Proving Grounds, Utica, Mich., has accepted a similar position with the Eaton Mfg. Co., Spring Division.

COL. FREDERICK C. HORNER has retired from General Motors Corp., where he was director, Field Operations Section, Distribution Staff, after 25 years service with the Corporation. He is returning to New York, where he will be active in consulting work touching upon transportation and allied problems. Horner served overseas during World War I and rejoined the Army early in World War II to organize and operate the Highway Division, Transporation Corps, AAF. He was SAE Vice-President for Transportation and Maintenance in 1930, was a Councillor in 1935 and 1936, and has continued active in SAE affairs.

Previously with the National Carbon Co., ROBERT S. MERITHEW is now an automotive engineer with A. E. Friedgen, Inc., in New York City.

PETER ALTMAN has been appointed vice-president of Continental Motors Corp., in charge of its newly organized Multi-Tool Division. He has been on Continental's staff for five years as a consultant and as chairman of its engineering council and will continue in these duties, making his headquarters in Detroit. In 1942 he was SAE Vice-President for Aircraft Engineering.

Recently promoted to director of engineering for all products of the Crane Packing Co. in Chicago, CARL E. SCHMITZ had been chief engineer of this company.

H.A. (HAL) LOCKWOOD, previously manager of the Protek-Plug Department of the Chandler-Evans Corp., has become associated with Clark Metal Products, Inc., Fairfield, Conn., as chief engineer in charge of dehydrator plugs, new machine design, and new products research. Very active on SAE committees during the war, Lockwood is the author of several papers on preservation through dehydration.

PAUL C. ROCHE has been elected national director of the Society of Plastics Engineers and treasurer of the Northwestern Pennsylvania Section of the SPE. Roche is sales engineer of NOSCO Plastics, Erie, Pa.

Recently appointed general manager of Morris Motors, Ltd., Engines Branch in Coventry, England HERBERT CLARK was with Morris Commercial Cars in Birmingham.

ARTHUR R. CONSTANTINE was recently appointed director of engineering at the Indian Motocycle Co.

A former general service manager of the Cummins Engine Co., Inc., Columbus, Ind., DAVID S. CRAVEN, has been appointed director of maintenance of the McLean Trucking Co., Inc., in Winston-Salem, N. C.

B. H. DeLONG, vice-president and technical director of the Research & Metallurgical Laboratories of the Carpenter Steel Co., Reading, Pa., has been awarded the Bradley Stoughton plaque. This plaque is presented to the Lehigh Valley metallurgist who has accomplished the most outstanding and generally useful metallurgical work. It is given by the Lehigh Valley Section of the American Society for Metals.

A. N. MORTON has been elected a vice-president of Mack Trucks, Inc. He is in charge of the company's manufacturing operations and will continue to make his headquarters in New York City.

Previously truck and bus engineer at Continental Motors Corp., PAUL W. FULLER

has become chief body engineer with the General American Aerocoach Co. in East Chicago, Ind. He had been chairman of the Western Michigan Section of the SAE.

Prior to becoming a project engineer with Kiekhaefer Aeromarine Motors, Inc. in Fond Du Lac, Wis., WALTER H. SCHACHENMAN was resident engineer for the Eclipse Machine Division in Elmira, N. Y.

Now a designer in confidential design at North American Aviation, Inc., Inglewood, Calif., DONALD J. NAUMANN was chief engineer for the Brown Aircraft Co. in Montebello, Calif.

Recently becoming president and general manager of Val H. Rhodes & Co., Memphis, Tenn., VAL H. RHODES was district sales manager for PurOlator Products, Inc., Newark, N. J., and also a field engineer on filtration for this company.

RICHARD B. HOOK, who had been a laboratory technician at Wright Aeronautical Corp., Paterson, N. J., is now a test engineer for the General Electric Co., Schenectady, N. Y.

No longer supervisor of Production Control, Maintenance Division, American Airlines, Inc., Tulsa, Okla., GEORGE W. GILMER, III, is a senior engineer, Powerplants, in the Service Division with this company, same city.

FRANK H. OLTON, now vice-president of Samuel Moore & Co., Mantua, Ohio, was formerly sales manager for this company.

Until recently a project engineer with the Eaton Mfg. Co., Wilcox-Rich Division inDetroit, ARTHUR W. HOLLAR, JR., is now sales engineer in the Heater Division of this company in Cleveland, Ohio. In 1947, Hollar was on the Promotion of Junior Activities Committee for the SAE Detroit Section.

Heretofore connected with the British American Oil Co. in Toronto, Ont., Canada, ALBERT B. RUPERT recently became sales manger for Consolidated Truck Lines, Ltd., same city.

Collaborating on the book, "Aluminum and Its Applica-

tions, by HIRAM BROWN were: ALEXANDER KLEMIN, M. H. YOUNG, N. E. WOLDMAN, R. B. MEARS, and R. A. FRYE. These and other materials specialists in various industrial fields have contributed to this treatise on aluminum technology. It deals with fundamental properties of aluminum alloys, their uses, and fabrication processes. Eight chapters of the book deal with special uses of aluminum in aircraft, automotive, railroad equipment, marine assembly, electrical appliance, and chemical manufacture. The last chapter describes progress with alumin today and forecasts rapi advances in many field tomorrow.

OBITUARIES

PATRICK J. FLAHERTY

Patrick Flaherty, chairma of the board of director of the Johnson Bronze Co., New Castle, Pa., died suddenl in Miami Beach on Feb. 12.

In 1909 he joined the American Car & Ship Hardware Co. as an accountant. In 191 he was asked by the Munci Products Co. if his compan could manufacture bron bearings for automobiles. Although the company neve had, Flaherty was prompte to say yes. He installe new machinery, threw o many lines the company w manufacturing and soon bron bushings and bearings wer being produced. This was th beginning of Johnson Bronze C He purchased the plant i 1928, and since then h been the principal owner.

MAURICE GOUDARD

One of the leading automotive engineers and manufacturers of France, Mauri Goudard died Jan. 17 in Par at the age of 66. Preside of the Societe Solex, a pas president of the Societe d Ingenieurs De L'Automobile, he had been a leader bot in the engineering and pro duction fields of the auto motive industry in his n tive land.

Joining the SAE in 1925, h made a number of busines trips to this country, a

was considered to be one of the ablest manufacturing executives in Europe.

CARLOS H. ALLEN

For the past 29 years in the Detroit office of New Departure, Division of General Motors Corp., Carlos H. Allen passed away on Feb. 8 at the age of 56. He was the dean of bearing application engineers specializing in automotive design.

He was graduated from the University of Michigan in 1914 and joined New Departure in 1919. He also served the Army in World War I. In 1921 he joined the SAE.

KENNETH N. COOK

Kenneth N. Cook, assistant sales manager of the Rollway Bearing Co., Inc., Syracuse, N. Y., passed away suddenly on March 5. He had been with the Rollway Bearing Co. since 1927.

New Members Qualified

These applicants qualified for admission to the Society between Feb. 10, 1948 and March 10, 1948. Grades of membership are: (M) Member; (A) Associate; (J) Junior; (Aff.) Affiliate; (SM) Service Member; (FM) Foreign Member.

Baltimore Section: Leroy W. Klimm, Jr. (J), William E. Monroe (J).

British Columbia Group: Kenneth Logan Hall (J).

Canadian Section: Charles

Edw. "Ted" Aylett (A), Lawrence M. Hart, Jr. (A).

Central Illinois Section: Kenton Carl Opperman (J).

Chicago Section: Earl G. Boudreau (M), Ralph Gerald Harmon (A), Charles H. Malstrom (A), Francis R. Rogers (J).

Cleveland Section: Oliver H. Bancroft (A), Harry B. Biersdorfer (M), LeRoy Albert Difford (J), Donald E. Harpfer (M), Robert L. Larson (J), Reino J. Niemela (M), Glenn W. Quaint, Jr. (J), William G. Raney (J), John J. Sibisan (J), Ellis C. Yale (A).

Colorado Group: Harold [illegible] Robinson (A).

Dayton Section: Lewis Richard Catt (J), Tommy J[illegible] McCuistion (J), Krell Ear[illegible] Spires, Jr., (J).

Detroit Section: Winston [illegible] Ambrose (J), Arley Conra[illegible] Basha (J), Joseph P. Bushe[illegible] (A), Wayne A. Carlson (J), Harvey Seldon Freeman (J), William Edgar Gore (M), Joseph J. Hochbein (A), Robert P. Hoehner (J), Don [illegible] Horton (J), William Charle[illegible] Kelly (J), Carl H. Meil[illegible] (J), Worth H. Percival (J), Emile Loyola Provost (J), Lawrence B. Rivard (M), Morton I. Weinberg (M), Arthur G. Winter (M).

Hawaii Section: Charles R. Kroeger (A), David Hyman Meyer (A), Henry E. Meyer (A), Hymie J. Meyer, Jr. (A), Marco M. Meyer (A), Harold F. Schaffner (A).

Indiana Section: Dale Allen Hasler (J), George Thomas Kuhn (J).

Kansas City Section: Thomas S. Cruttenden (J).

Metropolitan Section: Thomas Edward Bailey (J), Herman Horton Bement (J), David M. Denzer (J), William Hamilton, Jr. (A), William H. Hoops (J), Joseph J. Meehan (J), Albert O. Schmitt (J), Elliot L. Taylor (J).

Mid-Continent Section: Robert S. Hinds (A), Fred L. Manton (A), L.O. Meyer (J), Samuel Judson Rudd (J).

Milwaukee Section: Russell E. Baetke (J), Robert G. Krueger (M), Robert V. Wagner (J).

New England Section: David C. Bailey (M), Marshall J. McGuire (J).

Northern California Section: K.L. Kipp (J).

Northwest Section: William Dale Edwards (J), Floyd A. Irvin (A).

Oregon Section: Melvin LeRoy Richey (A).

Philadelphia Section: William David Harshbarger (J), Charles B. Weiler (M).

Pittsburgh Section: Arthur E. Slagle (J), Keith William Stickle (J).

St. Louis Section: George L. Doelling (M), Victor Millman (J).

San Diego Section: Hamilton Laurie Stone (A).

Southern California Section: Weldon Beck Allbaugh (J), Don D. Doty (J), C. A. Forter, Jr. (M), Truman Gray Geddes (M), Richard Hulse (A), Alfred R. Kelly (A), Herbert T. Lund (J), John F. Maggio (J), William Frederick Roberts (J), Leonard Goodwin Walls (A), William Lindsay Westfall (M), Marshall C. Wood, Jr. (M).

Southern New England Section: E. Marston Moffatt (J), Alton J. Parks (J), Robert H. Webster (J).

Spokane Intermountain Section: C.C. Johnson (M).

Texas Section: Frank M. Jobes (M).

Twin City Section: C.E. Gobeil (A), Gordon Volkenant (M).

Virginia Section: William Berle Hubbard (J), Burton Howard Knight (A).

Washington Section: Stanley R. Darcey (A), Bernard Goldberg (J), Raymond W. Pecaut (SM), Francis J. Reilly (A), Carl C. Saal (SM).

Williamsport Group: Paul A. Spencer (A).

Outside of Section Territory: Herbert Massie Bradbury (J), J. H. W. Conklin (A), Ensign John Richard High (J), Robert Brewster James (M), Robert Carson Johnson (J), Quay Lyle (A), Leslie Moore Proctor (A), Ben H. Tyler (A), John M. Yantis (M).

Foreign: Jacques Allier (FM), France; Norman Edge Bassett (FM), England; Stanley John Breeze (FM), England; Richard Neville Clark (A), England; Prof. Antonio Fessia, (FM), Italy; Dante Giacosa (FM), Italy; Dr. Henry Lewis G (M), England; Joakim Ihle (FM), Norway; William Henry Slack (FM), England; John Watson Thorburn (FM), England; George Alfred Turner (FM), England; Major G. V Twist (A), Holland.

Applications Received

he applications for membership received between Feb. 10, 1948 and March 10, 1948 are listed below.

Baltimore Section: Earnest L. Kerr.

British Columbia Group: Jack Charles Ames, John Malcolm Billingsley, Nelson Howard Charlton, Walter Bailey Beaville, Clarence C. McAllister.

Canadian Section: Vincent Leo Boland, Clayton W. Squelch.

Chicago Section: C. O. Benson, Jacob B. Courshon, Jr., Otto Drolshagen, Mahesh Govind, R. J. Hicks, William O. Johnson, Otto R. Last, Ray C. Nelson, Thomas E. Sullivan, E. R. Walter, Sanford . Wanner, John Donald Warus.

Cincinnati Section: Arthur alter Lang, L. T. Ten Eyck.

Cleveland Section: Roy S. odbey, Donald J. Michel, loyd Jackson Moulton, William H. Robbins, William H. Thomas, Jr.

Colorado Group: Allan P. helly.

Dayton Section: Richard McKinney Allison, Morris man, Lamar T. Cox, William . Webster.

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